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Development of Advanced In-Cylinder Components and Tribological Systems for Low Heat Rejection Diesel Engines

Phase 1 Final Report

D.H. Reichenbach, K.L. Hoag,
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ORIGINAL CONTAINS
COLOR ILLUSTRATIONS

TABLE OF CONTENTS

List of Figures

List of Figures Cont'd

List of Figures Cont'd

List of Tables

SUMMARY

Commercial heavy duty diesel truck engines for the year 2000 will present engineering challenges in the areas of high temperature components and lubrication in the quest for lower fuel consumption and emissions. High specific output engines with reduced cooling for low heat rejection will induce thermal loading of the in-cylinder components beyond the levels of current engines. Pollutant emission concerns will continue to dictate many engine design details. Reduced generation and emission of particulates and oxides of nitrogen must be an integral part of any future diesel engine concept. Reduction of lubricating oil consumption by the cylinder kit is key to meeting future particulate standards.

The objective of the Cummins In-Cylinder Components Program is to identify and develop advanced in-cylinder components and tribological systems for application in future low heat rejection diesel engines. Phase 1 determined the appropriate engine configuration by conceptualizing, analyzing, and evaluating potential component alternatives.

In order to achieve the Heavy Duty Transport Technology (HDTT) program goal of .250 lb/bhp-hr brake specific fuel consumption, an engine system was proposed consisting of two stage turbocharging (to enable high BMEP at low engine speed), turbocompounding, low intake manifold temperature, low heat rejection, and high fuel injection and peak cylinder pressures. All components and systems are based on the Cummins L10 engine, a production engine of 10 litre displacement of modern design and proven performance. Thermal analysis of the cylinder head and liner determined that by using strategically placed oil cooling with suitable enhancement, it is possible to achieve acceptable component temperatures and thermal fatigue life while significantly reducing the overall system size and complexity by eliminating cooling water. A cylinder head concept using oil cooled grey iron, cast-in ceramic intake and exhaust ports, a high temperature metal combustion face insert, and plasma sprayed insulation on part of the combustion face is being pursued. The key in-cylinder component which has received much attention in this first phase is the piston. The concept being refined consists of a spherical piston/connecting rod joint for high load carrying capabilities and uniform thermal loads, ceramic fiber reinforced aluminum for light weight and high strength, and a combustion chamber thermal barrier coating of plasma sprayed zirconia.

I. INTRODUCTION

A. PROGRAM GOALS

This program is directed toward powerplant design for future heavy duty transportation requirements. The North American highway trucking industry of the year 2000 is the target of this technological development. As part of the Department of Energy's Heavy Duty Transport Technology (HDTT) program, the In-Cylinder Components effort shares the program goals of low fuel consumption (.250 lb/bhp-hr BSFC), low emissions as defined by the standards applicable to the expected production year, improved durability and reliability, and cost effectiveness. Realizing that meeting these goals requires a total system approach, some definition of the total system is required before in-cylinder component design and development can take place. As the HDTT goals are similar to goals Cummins Engine Company has for its future products, much of the system definition has already been done and is in place in the Cummins long term product plan. For obvious business reasons, the details of that plan are not discussed here, but the general features are covered by the concepts and designs of this program. To assist in the system definition and development of boundary conditions for detailed component analysis, cycle simulation using the Cummins code TRANSENG was performed and resulted in a prediction of .250 lb/bhp-hr BSFC while suitably constraining injection timing to reflect a low NOx engine.

Emissions standards are not in place for the year 2000 which is the time frame for production of concepts developed in this program. For the purposes of concept selection, the assumed 1994 limits of 5.0 g/bhp-hr NOx and 0.1 g/bhp-hr particulates were used (the 1994 NOx limit has not yet been established). Concepts that lend themselves to even lower levels were given preference.

While meeting performance and emissions targets requires a system solution, the primary roles are performed by the fuel injection and air induction systems in a diesel engine. The in-cylinder components define the geometry and heat transfer of the combustion chamber and play a very important secondary role. All components must be consistent with the system requirements for cylinder pressure, temperature, frictional losses, etc. It is here with the in-cylinder components that the emphasis of this program lies. The future trends in the engine system are generally known and analysis performed on this program further defined the demands on the in-cylinder components. Developing pistons, rings, cylinder liners, and cylinder heads consistent with the total system requirements of the year 2000 is the object of this program.

Durability targets appropriate for the year 2000 include improvements over current production engines which average from 400,000 to 700,000 miles before the first overhaul. A target of 1,000,000 mile durability was chosen reflecting the customer expectation of continual improvement.

B. BACKGROUND INFORMATION

Future diesel engine design is dictated by customer expectations, environmental requirements, market demands, and competitive pressures. While much of the current industry effort is directed towards exhaust emissions reductions, most gains in this area are transparent to the end user. Users of heavy duty diesels are becoming more sophisticated in their approach to trucking and becoming more demanding of their engines. Each improvement from the engine manufacturers in the areas of transient response, fuel consumption, oil consumption, or durability is met by increased customer expectations. Fuel costs will continue to dominate the cost structure of the vast majority of trucking concerns so the informed customer will demand continued improvement in fuel consumption. Matching of drivetrain components and controls to the duty cycle will allow engines to spend much of their time in efficient low rotational speed operation. This will require high torque and fast response at low engine speeds to allow good drivability during transitions between speed and load regimes.

Deregulation of the trucking industry has resulted in an increase in the average gross combined weight of highway trucks. Vehicle and tire improvements are resulting in large reductions in rolling and wind resistance. An interesting phenomenon has resulted in that the horsepower to maintain a steady speed on level highways is being reduced but the horsepower required to climb grades and meet customer transient response expectations is increasing. Figure 1 demonstrates this effect. At a typical vehicle weight, the level road power requirement will decrease from 200 to 140 horsepower between the 1980's and the year 2000.

This power requirement reduction is immediately reversed if even slight improvements in gradability of less than 1% grade are desired. Vehicle acceleration follows the same trends. The mass of the vehicle and power available will continue to dictate vehicle response. F=ma (force equals mass times acceleration) will continue to be valid. The engine which meets this demand has relatively low displacement and component size for low friction and good fuel economy during cruising conditions, but has the capability of delivering high power when required for acceleration or climbing grades. Vehicle designs for improved aerodynamics give preference to compact engines and cooling systems. These are some of the factors making a small, high output engine attractive in the future.

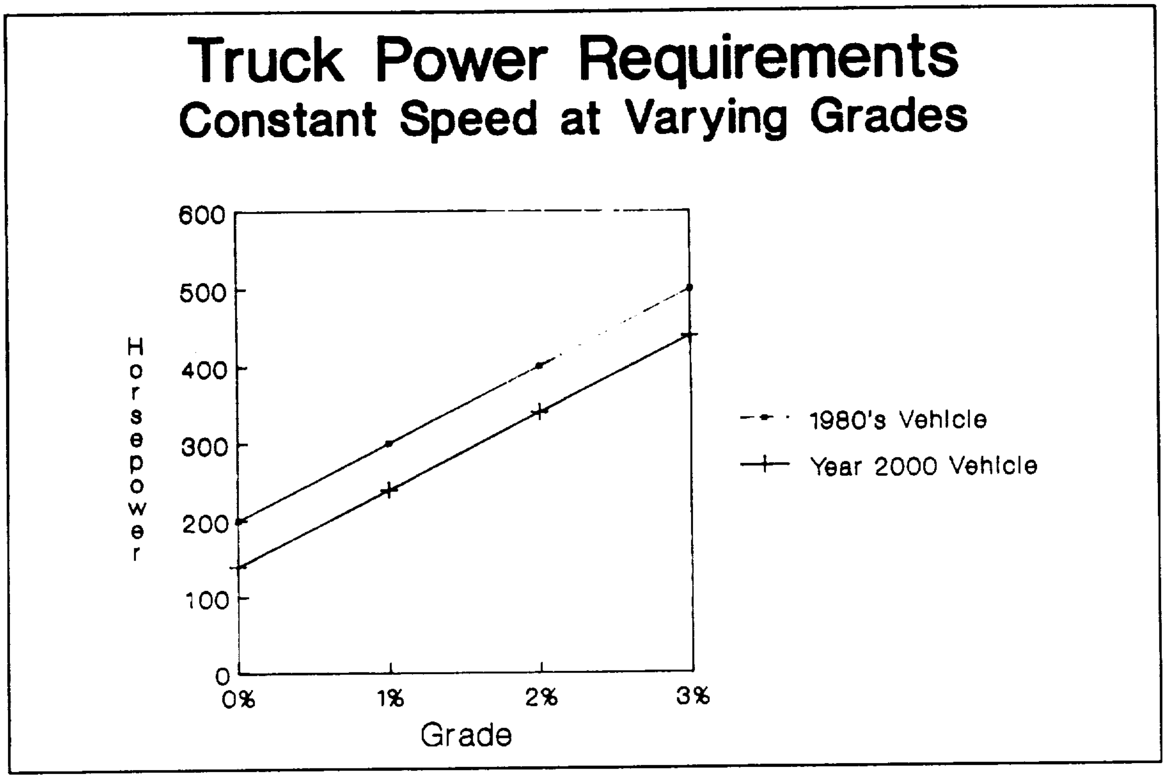


Figure 1

The basis for all component design and analysis on this program is the Cummins L10 engine. This is representative of state of the art diesel engines. The L10 has a 125 mm bore and a 136 mm stroke for 10 litres displacement. It is currently sold in turbocharged and aftercooled configuration for highway truck application at ratings from 240 to 300 horsepower. The combustion system utilizes high pressure unit injectors and low swirl (considered quiescent) combustion chamber. While compact in overall size, the L10 is designed with large crankshaft, bearings, and connecting rod for good durability at high cylinder pressures and output levels. Since all new components developed on this program will fit into the L10, test engines will be readily available and baseline data for comparison will be easily obtained.

C. GENERAL CONCEPTS STUDIED

Our approach to defining the concepts to be developed started off by envisioning all of the features that might be included in an advanced heavy duty transport engine. We then went through a process of narrowing down the options by evaluating the potential of each concept. Some of the evaluation criteria were:

- Emissions and fuel consumption potential
- Durability potential
- Capability of increased power output
- Reduced heat rejection
- Cost effectiveness
- Year 2000 technology

One important idea is that of appropriate technological risk. We are striving to make large advances in engine technology but at the same time want to constrain ourselves to that which is feasible for introduction into a production engine in the next 10 years. It is also our hope that there will be spin offs of the technology developed on this program that will be production ready in the 1990's.

There is a great deal of interest in alternate fuels from both strategic resources and emissions standpoints. We have chosen to concentrate on diesel fuel due to its availability and high BTU content. Certain concepts such as combustion chamber insulation do make the use of a wider range of fuels feasible and are included. The heat release rate shapes used in the cycle simulations and the thermal boundary conditions used for component analysis are based on diesel combustion.

Exhaust energy recovery is an important part of most advanced engine concepts. A Rankine bottoming cycle was investigated as having the potential for excellent use of the available energy. Using the available literature and internal studies at Cummins it was determined that such a system would not be cost effective in the time frame addressed in this program. Turbocompounding was chosen as an appropriate method for extracting useful work from the exhaust. Turbocompound systems have been proven on a prototype basis by Cummins and other engine manufacturers. The hurdles to clear before turbocompounding becomes a reality are aerodynamic and mechanical efficiency, back pressure increase, and cost.

An overhead camshaft engine design was considered to enable better valve dynamics, increase fuel injection pressure capability, and reduce engine complexity. This technology is not new and can be seen in production on many gasoline engines and a limited number of diesel engines. The current L10 engine has the camshaft mounted very high in the block for many of the performance benefits derived from an overhead cam. It was determined that development of an overhead camshaft system would not be a significant step technically and that resources would be better applied to the development of other components and systems.

Certain modifications to the normal diesel cycle were considered. These included variable valve timing, variable compression ratio, and modified volume schedule (non slider-crank mechanism).

Variable valve timing is technically achievable and is receiving much attention in naturally aspirated gasoline engine research. The benefits of variable valve timing are fairly moderate on a turbocharged diesel engine and outside a narrowly interpreted scope of "in-cylinder components" and as such were not dealt with further. Both variable compression ratio and modified volume schedule require drastically different engine designs from current diesel engines. Considerable expense and engineering time would be required to prove any such concept and would detract from developments in other components. The potential performance improvements were judged to be moderate and do not warrant further study in this program.

It was determined that in order to obtain the greatest amount of relevant engine technology development, this program should concentrate on a turbocharged and turbocompound version of the L10 engine. The specific components to receive detailed design and development include the piston, piston rings, cylinder liner, and cylinder head.

II. PERFORMANCE ANALYSIS

Since one of the primary goals of the Heavy Duty Transport Technology program and in turn the In-Cylinder Components program is reducing fuel consumption, predictions of how that reduction might be accomplished are important to provide program direction.

Early in the program a study was undertaken using previous unpublished work and some straightforward calculations to estimate what improvements would be necessary to achieve .250 lb/bhp-hr BSFC. Using .320 BSFC as a baseline (some current L10 engines actually perform better than this), a 22% improvement is necessary to reach .250 lb/bhp-hr. Figure 2 shows in pie chart form where the predicted improvements are expected to be found. The largest improvement is from the combination of insulation and turbocompounding. Insulation of the combustion chamber and exhaust passages accompanied by waste heat recovery by turbocompounding account for a 6% improvement in fuel consumption. Further aerodynamic and mechanical efficiency improvements can increase this another 1.5%. Increasing the efficiency of the turbocharger compressor, turbine, and bearing system will result in 2% fuel consumption improvement. Parasitic loss reduction is addressed in three ways. Rubbing friction reduction from cylinder kit design will give 1% BSFC reduction. Reducing the engine rpm at vehicle cruising speeds will result in a 3% reduction. Reduced parasitic loads related to the cooling system (pumps and fan) will give another 2%. Improvements related to the actual combustion process are lumped into three areas. Increased peak cylinder pressure will give 2% improvement, combustion chamber optimization and crevice volume reduction will account for 2%, and high injection pressure fuel system will give 2.5%. While not based on rigorous analysis in

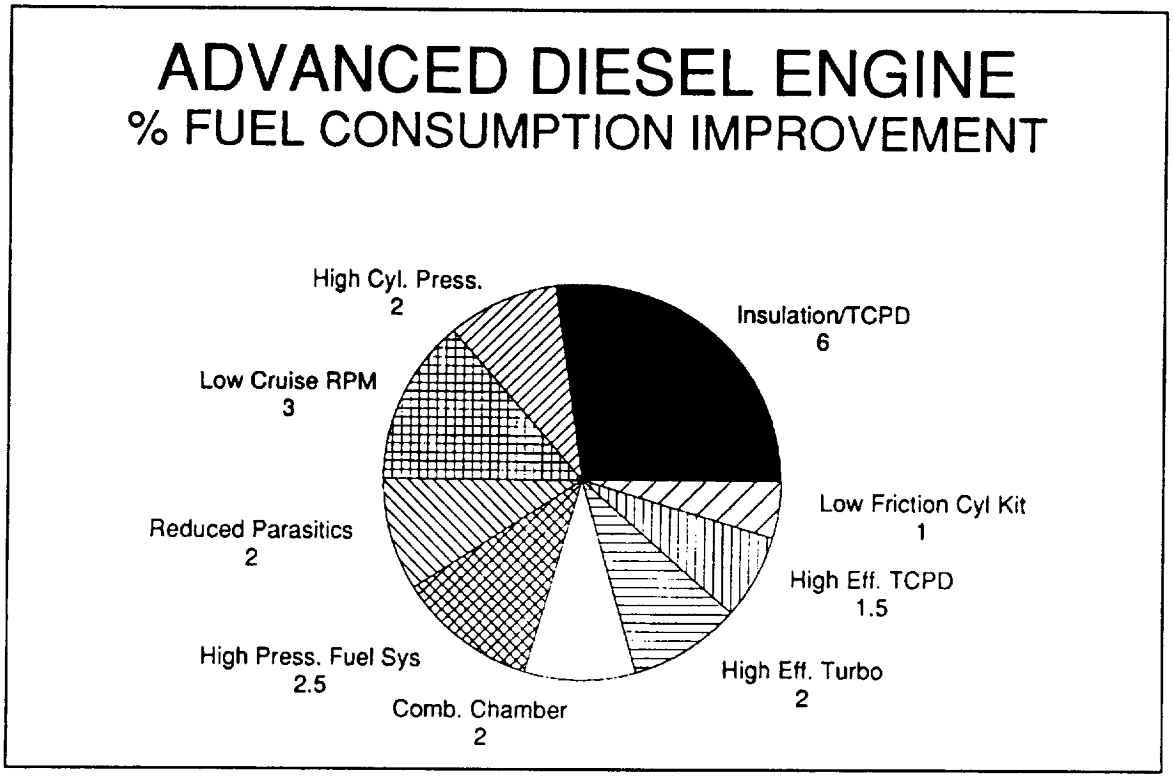


Figure 2

all cases, these predictions are felt to be representative of good engineering judgement and should be reasonably accurate.

A more detailed analysis of areas for fuel consumption improvement was performed using the Cummins diesel cycle simulation code TRANSENG. The original code was developed at Imperial College by Dr. Neil Watson. Cummins purchased the code in 1978 and since that time have made numerous developments and improvements in the code while still retaining the original "TRANSENG" name.

The reason for this more detailed cycle simulation was to better define the expected environment for the in-cylinder components. In-cylinder components are the heart of the diesel engine powerplant system foreseen for the year 2000. This effort helped generate the detailed boundary conditions needed in order to perform the necessary thermal and mechanical analyses of the components involved in this program. These typically are in the form of temperatures, pressures, and heat transfer coefficients.

A parametric study was undertaken to evaluate various options available to improve fuel consumption. As a baseline, an L10 engine such as might be appropriate for 1991 was used. Suitable engine test data was available for model "calibration". The chosen baseline rating was 320 HP @ 1800 rpm. Injection timing was constrained to the range appropriate for an engine producing 4 to 5 g/bhp-hr NOx to keep emissions in focus. The investigated variables were:

Horsepower/BMEP
Engine RPM
Turbocompounding
Power turbine pressure ratio
Turbocompound system efficiency
Single and 2 stage turbocharging
Turbomachinery efficiency
Compression ratio (reciprocator)
Fuel injection timing
Peak cylinder pressure
Combustion chamber insulation
Air/fuel ratio
Intake manifold temperature

Since the investigations were done in a parametric fashion, the magnitude of change resulting from each parameter variation is more important than the absolute value of BSFC on each of the accompanying plots. The baseline configuration was allowed to change periodically during the analysis process to reflect modifications suggested by the results to date. As a final check, an optimum configuration was constructed with a resultant predicted BSFC of .250 lb/bhp-hr.

Increasing the fueling and air flow of an engine will increase the output approximately linearly while friction increases at a much lower rate, thereby improving BSFC. Peak cylinder pressure will also increase. Assumptions have to be made that the combustion characteristics will remain predictable as fueling is increased, i.e. proper fuel/air mixing will continue to exist, fuel impingement on the cylinder walls will not have adverse effects, etc. Figure 3 plots out these interrelated effects. The higher the output, the more the demand on both the fuel and air handling systems. All of these analyses assume that suitable systems are available to deliver the fuel and air. In reality, much development would be needed in both the fuel and the air handling systems to deliver the complete powerplants illustrated by these studies.

An effective way of reducing friction to improve BSFC is reducing engine operating rpm. Figure 4 demonstrates what happens to a turbocharged engine when power is held constant but speed is varied. Lower rpm will continue to give improved BSFC until there is insufficient air for proper combustion. Subsequent rematching of the turbocharger can provide additional air at low speeds at the expense of high speed operation. Variable geometry turbochargers can provide sufficient air for low engine speeds

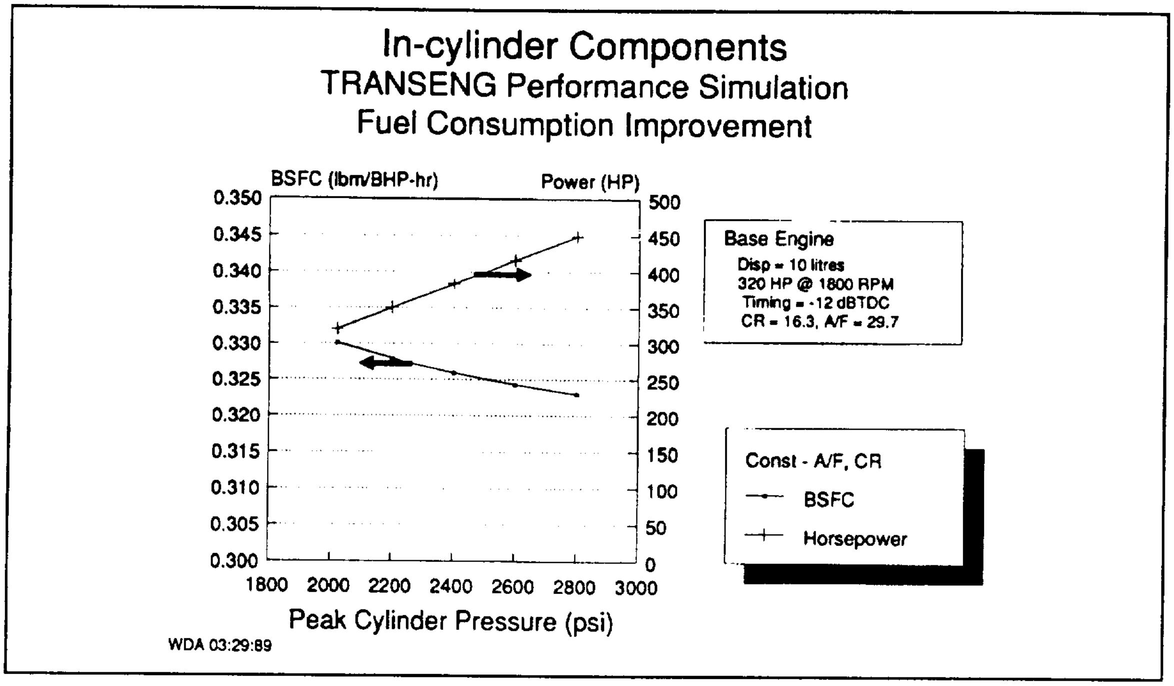


Figure 3

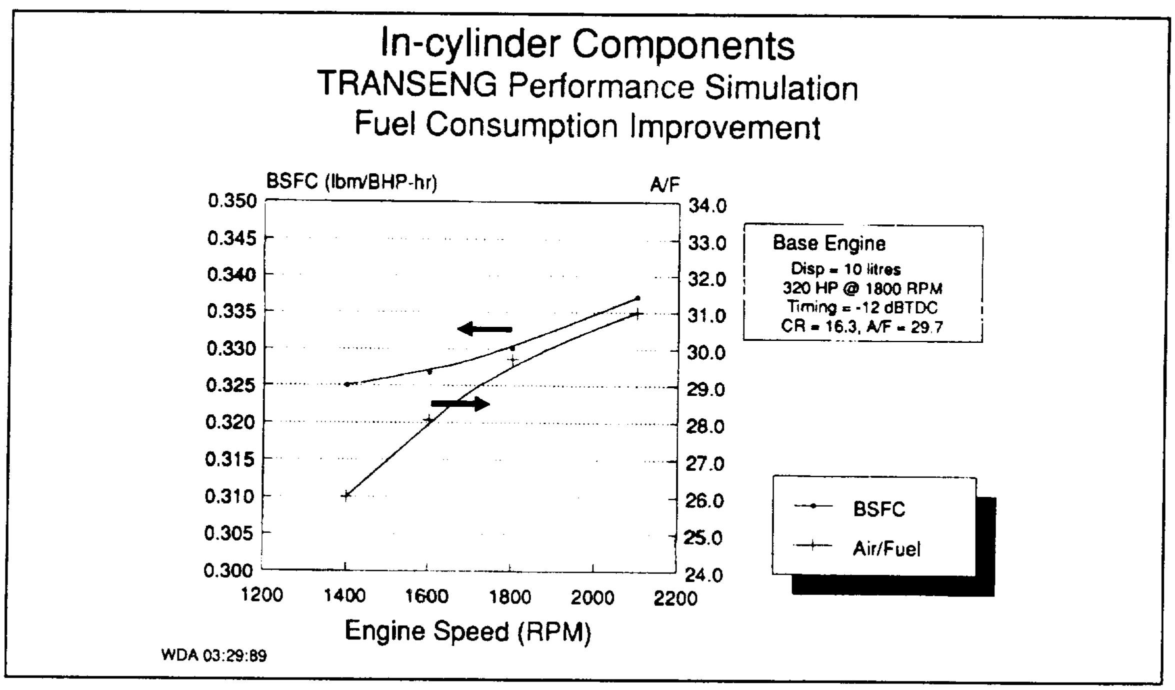


Figure 4

while still providing the capability of broad speed range operation which is necessary for drivability in transient situations.

Turbocompounding is an effective way of using waste heat in the exhaust stream to generate useful power. However, adding a power turbine to the exhaust increases the back pressure to the engine and results in additional work being required to pump the air through the engine. Pressure (or expansion) ratio of the power turbine is an indicator of both the amount of work obtained and the additional restriction on the air flow since the power turbine exhausts to atmospheric pressure. Figure 5 shows that there is an optimum pressure ratio for a given set of engine parameters. This plot was made assuming a combined aerodynamic and mechanical efficiency of 85% in the turbocompound system. Increasing the efficiency to 90% changed the optimum pressure ratio from 1.5 to 1.6. Turbocompounding will account for roughly 10% of the total engine output at the optimum balance of reciprocator versus power turbine.

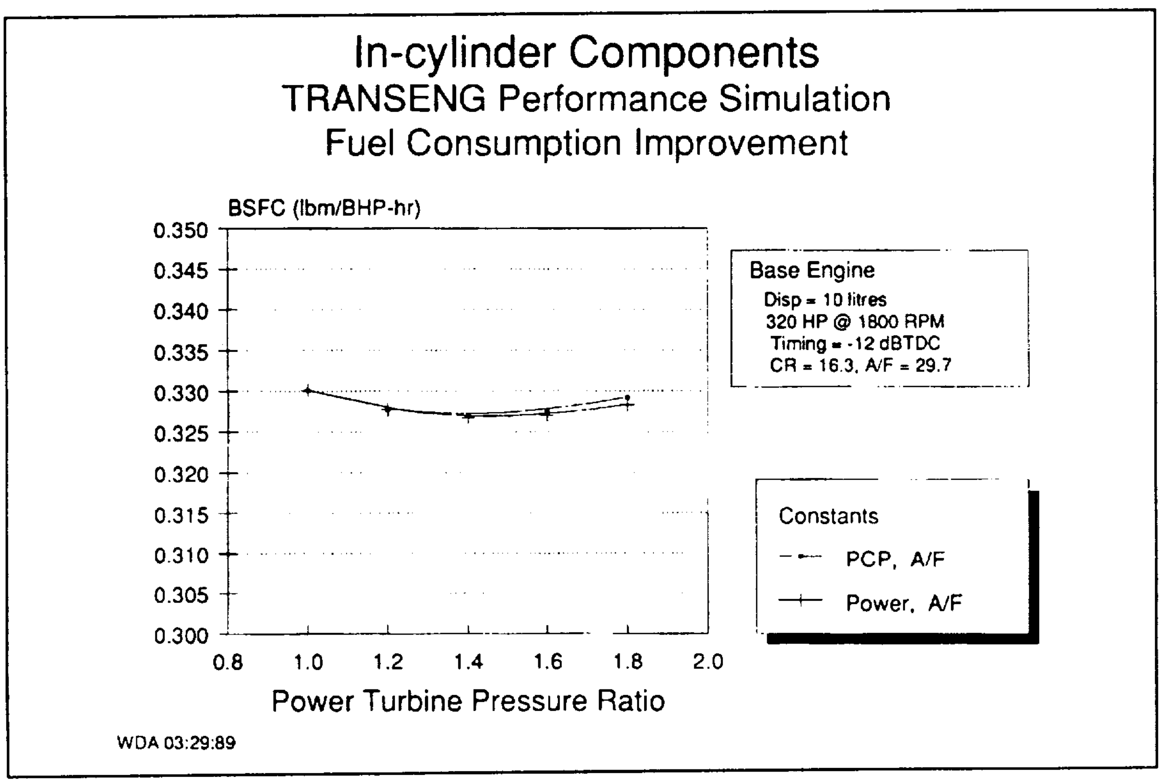


Figure 5

A portion of the compression done on the engine working fluid (air) is done external to the cylinder in the turbocharger. Turbocharger efficiency is therefore important to overall powerplant efficiency or BSFC. Figure 6 shows the system benefit of improved turbocharger efficiency. The combined turbocharger efficiency shown is the product of compressor efficiency and turbine efficiency. In this study, both were considered to be equal for simplicity of calculation. The baseline 56% overall efficiency reflects 75% compressor and turbine efficiency. The maximum 72% combined efficiency reflects 85% efficiency of the compressor and turbine.

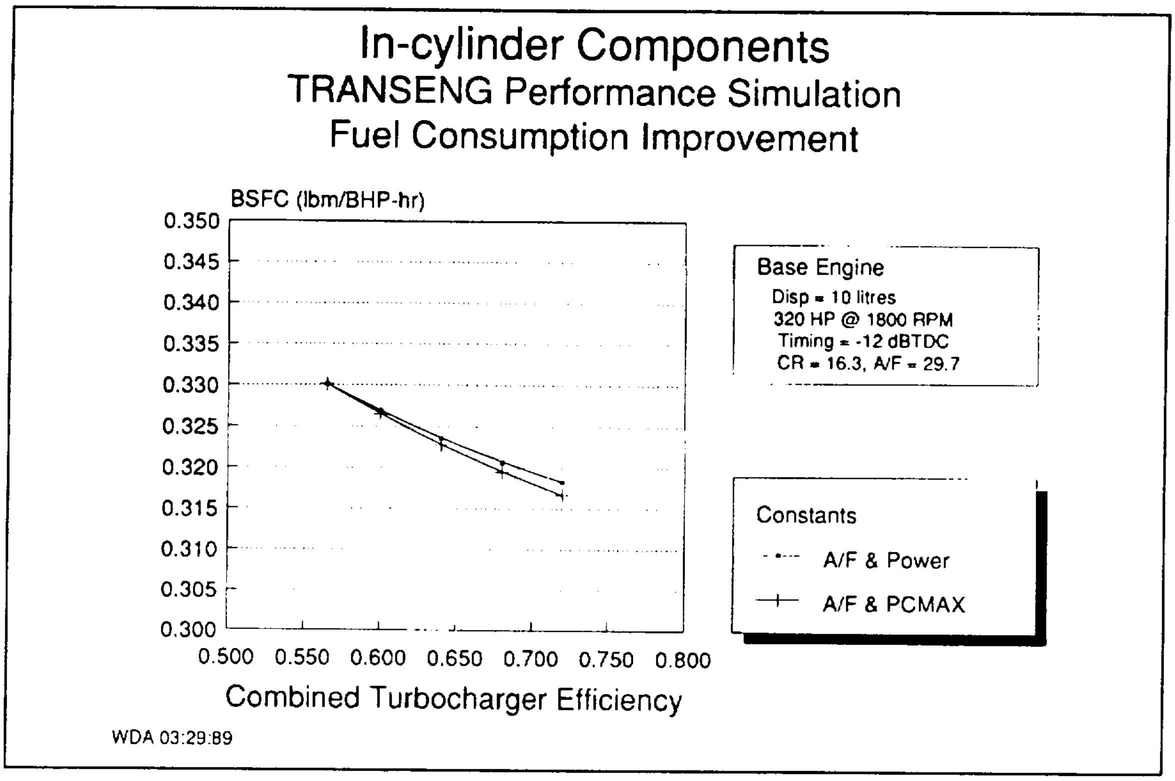


Figure 6

Figures 7 and 8 demonstrate the combined effects of changing compression ratio and fuel injection timing at both 400 and 475 horsepower output. At a constant power output, increased compression ratio will increase the peak cylinder pressure and the cycle efficiency. A practical upper limit to compression ratio occurs when the combustion chamber becomes too small at top dead center to enable good combustion without excessive wall impingement. The practical upper limit for high output quiescent chamber engines is estimated to be around 18:1. Each engine configuration and operating condition will also give an optimum injection timing for best fuel consumption. The two

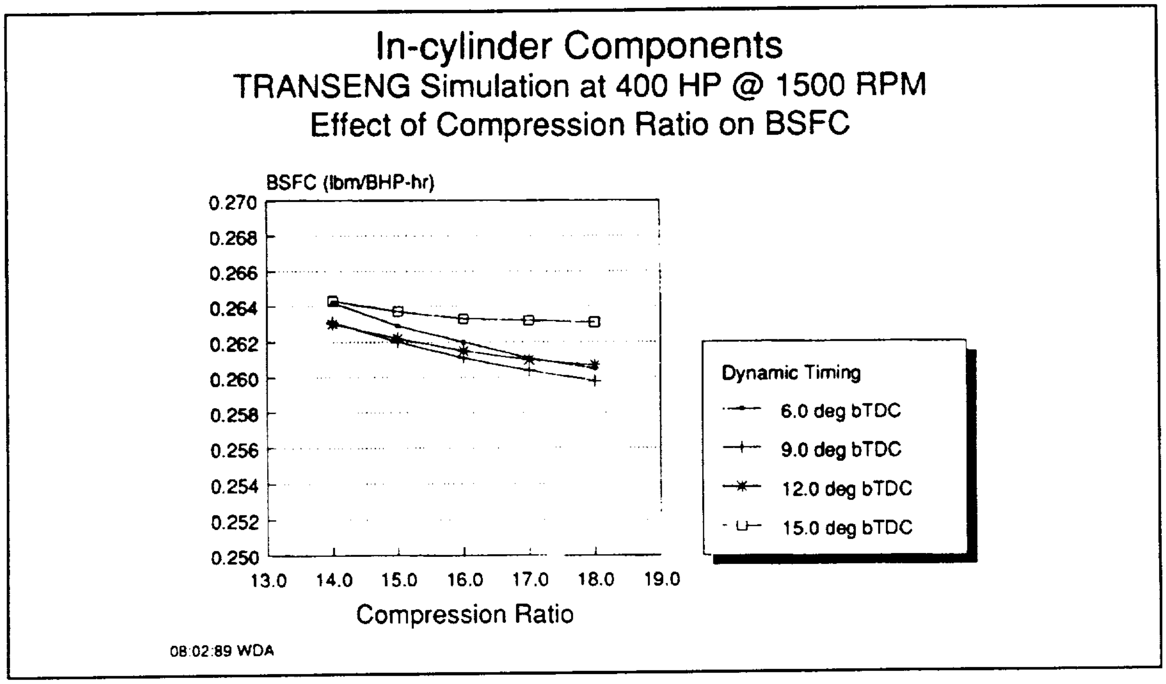


Figure 7

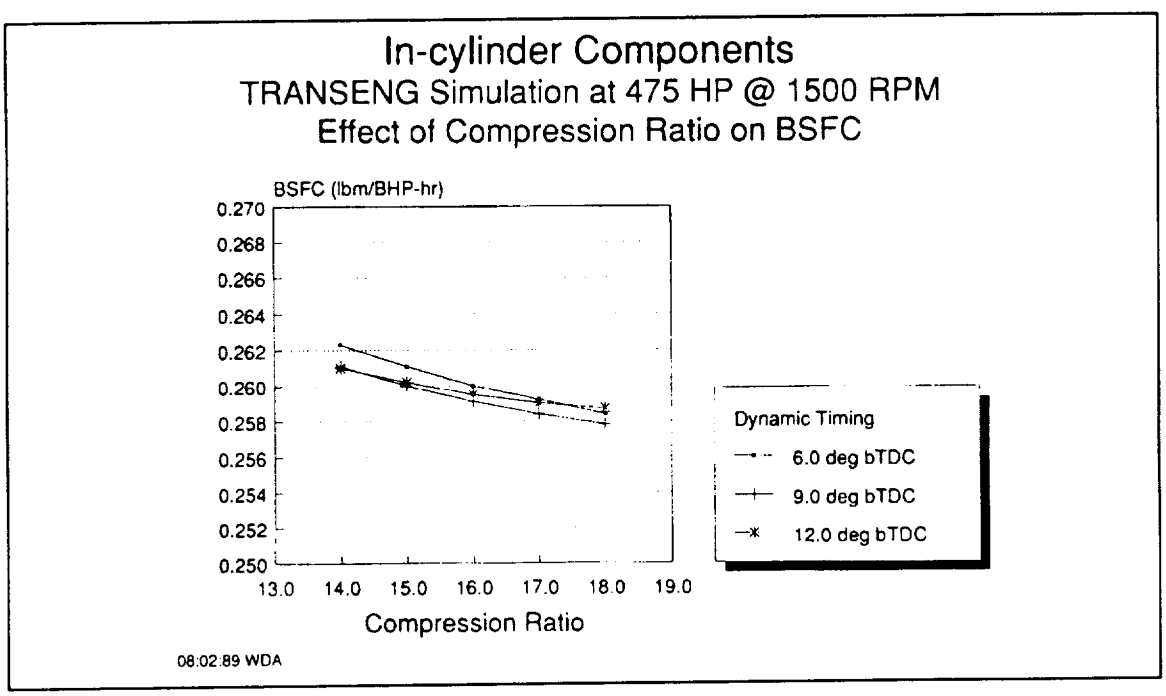


Figure 8

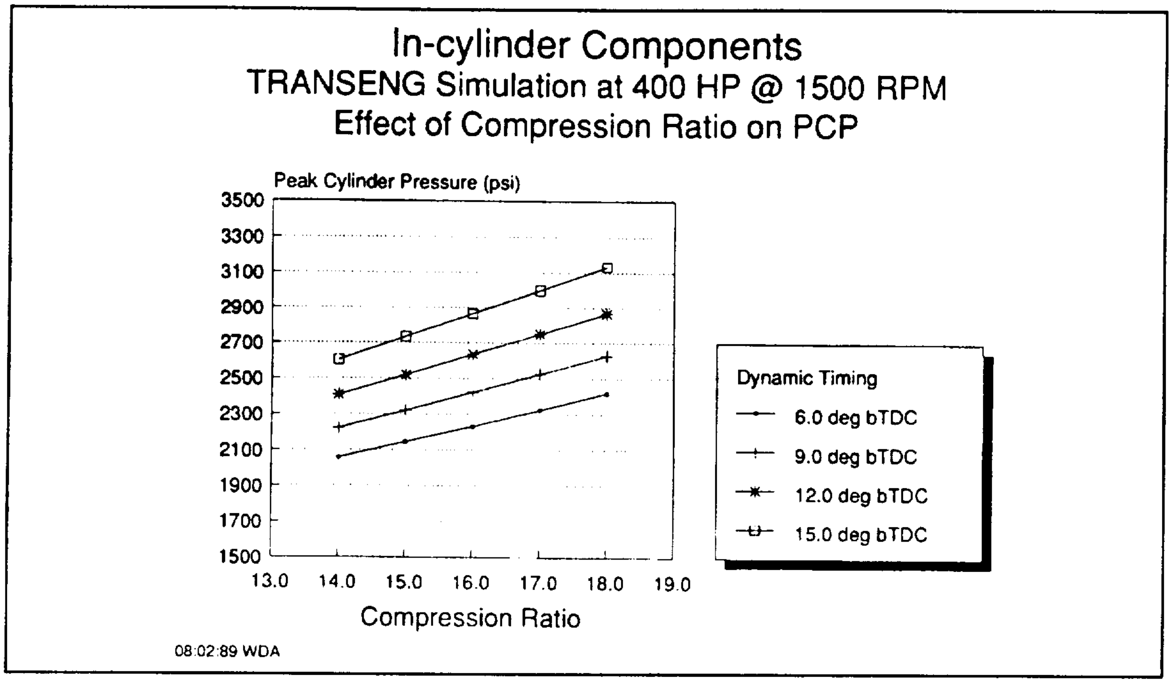


Figure 9

configurations modeled here had an optimum timing of around 9 degrees before top dead center. Peak cylinder pressure will continue to increase with injection timing as shown in Figure 9, even though BSFC is no longer improving. Consideration of NOx generation will also keep the more advanced timings from being practical.

Heat transfer from the working fluid to and from the walls of the combustion chamber is an area of great interest because of the amount of fuel energy lost by heat rejection to the coolant. It is also an area of many unknowns such as: how is heat flux to a hot wall affected by the approaching flame, what is the effect of varying wall material thermal diffusivities, and what are the combined emissions and performance effects? In this study we used rather simplistic models to predict the effects of thermal insulation. The model was modified in such a way as to decrease the effects of volumetric efficiency loss due to intake heating.

This was justified because of the decision to insulate the intake ports (see later section on proposed concept). As seen in Figure 10, the model predicted approximately 6% improvement in fuel consumption with a 40% reduction in in-cylinder heat transfer. The model also assumed no significant changes in heat release rate due to the presence of hot walls.

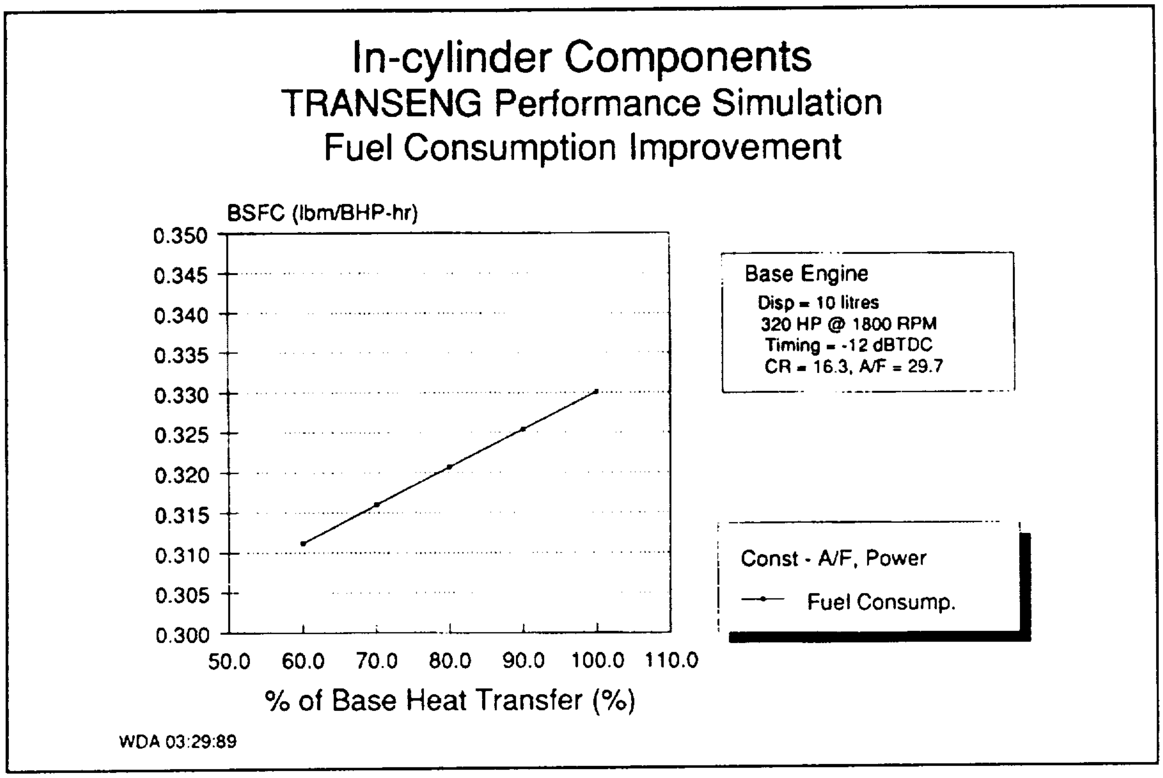


Figure 10

Air fuel ratio effects are demonstrated in Figures 11 and 12. At a fixed power and compression ratio, air/fuel ratio is changed by changing the boost pressure. Fuel consumption improves and cylinder pressure increases with increasing boost. Again, the assumption was made that advanced turbomachinery would be available to supply the required air flow as indicated by the model. Lowering intake manifold temperature is an effective means of both improving fuel consumption and reducing NOx emissions through lower in-cylinder gas temperatures. Through advanced intercooling schemes or other means it is possible to achieve very low actual or effective intake temperatures. These systems might include refrigerant systems, charge air expanders, alcohol fumigation, or early intake valve closing (Miller cycle). The benefits that could be expected are shown in Figure 13. While the BSFC improvement is rather small, the emissions benefit would be quite large if currently observed trends could be extrapolated to lower temperatures.

A combination of parameters reflecting major improvements throughout the system was used to model a .250 lb/bhp-hr BSFC engine. These are tabulated in Figure 14 along with a slightly less technically aggressive set of parameters that resulted in a predicted .270 BSFC. As in all the above referenced

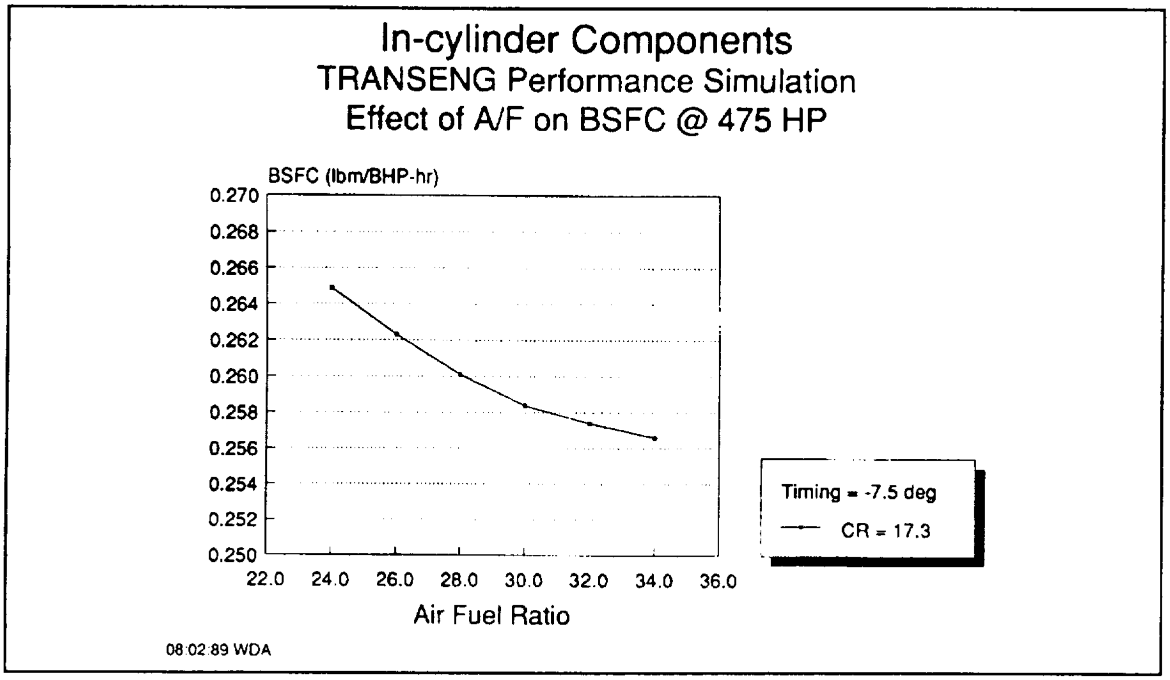


Figure 11

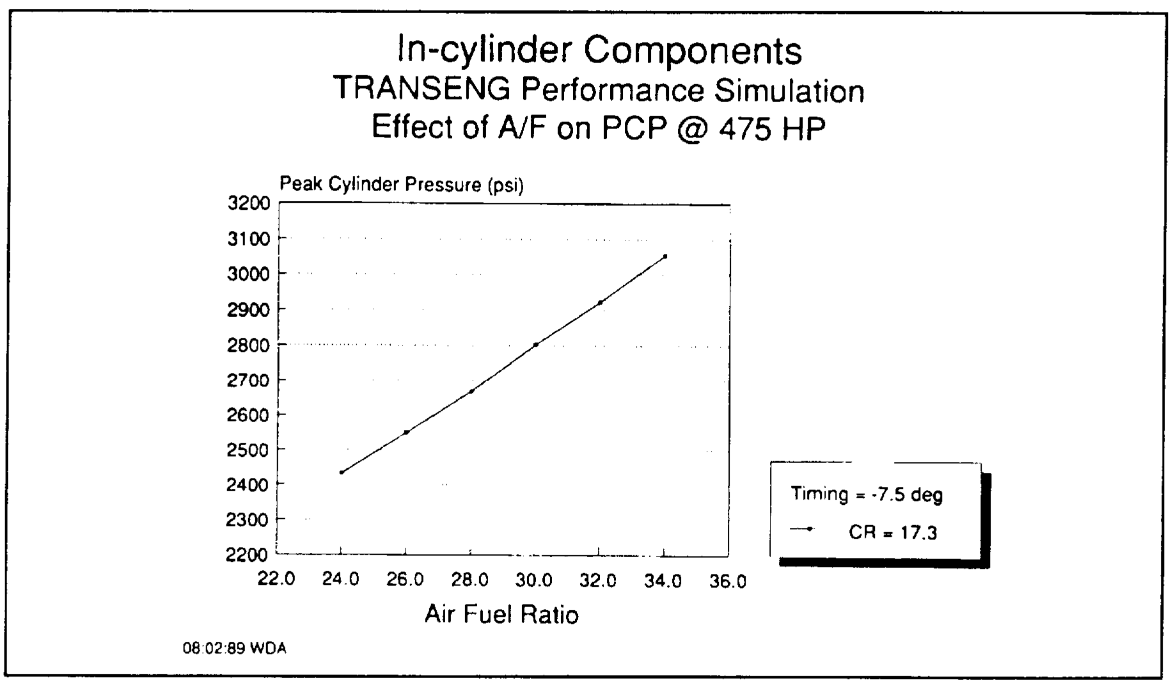


Figure 12

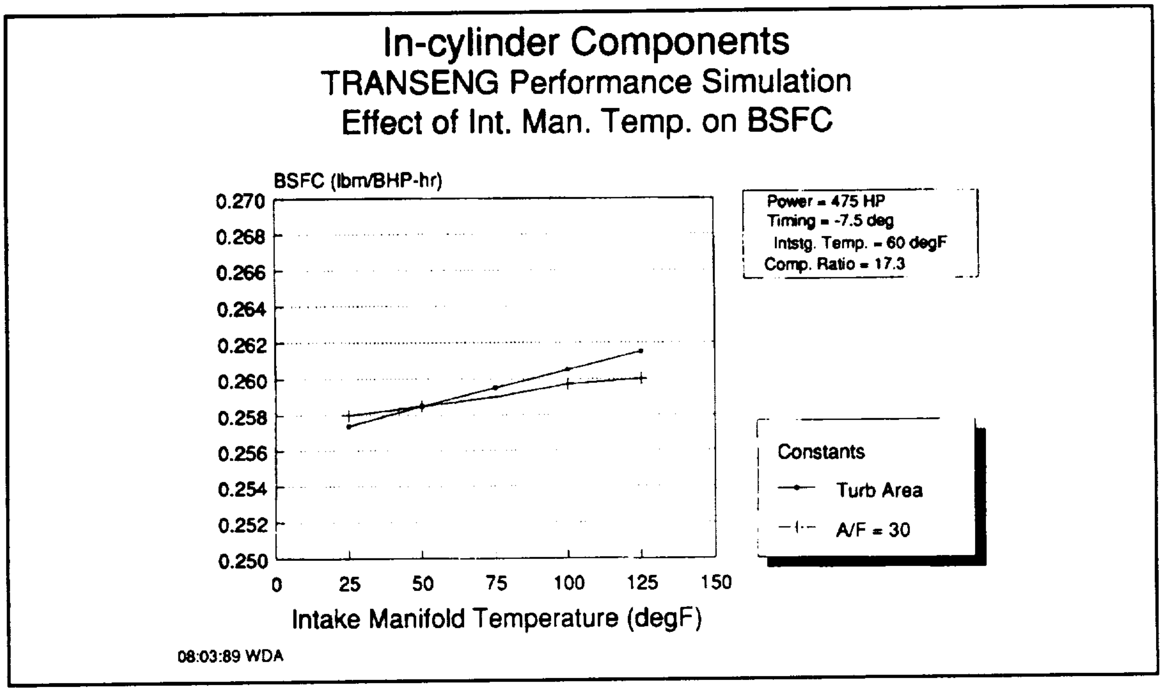


Figure 13

Engine Configurations for Achieving Low BSFC Simulation Results			ΔBSFC
BSFC (lb/bhp-hr)	.270	.250	.020
Engine Speed (rpm)	1600	1500	.001
Power (HP)	435	472	
BMEP (psi)	352	410	.003
Air Fuel Ratio	28.0	30.0	
Peak Cylinder Pressure (psi)	2500	2800	.002
Compression Ratio	16.3	17.3	
Dynamic Timing (deg BTDC)	12.0	7.5	
Combined Turbo Efficiency (LP)	0.72	0.75	.005
Combined Turbo Efficiency (HP)	N/A	0.75	
Interstage Temperature (deg F)	N/A	60	
Intake Manifold Temp. (deg F)	105	50	.002
Power Turbine Pressure Ratio	1.50	1.60	.0005
Power Turbine Efficiency	0.85	0.90	.0015
Reduce Heat Transfer by (%)	40	40	.000
Reduce HRR Duration by (%)	0	18	.004
Reduce Engine Friction by (psi)	6.0	7.0	.001

Figure 14

calculations, displacement was 10 litres. One parameter that was not mentioned above is the heat release duration. The heat release duration as normally reflected in the model is based on currently used fuel systems and injection pressures. For the advanced system model, it was assumed that by use of advanced high pressure fuel injection, the heat release rate duration would be shortened due to better fuel/air mixing. The engine friction reduction (FMEP) is due to a combination of reduced parasitics in the cooling system and low friction cylinder kit designs. This is not the only combination of engine parameters that would have a predicted .250 BSFC and any future demonstration at that level would likely differ in some aspects, but this is a possible approach to very low fuel consumption with a sound technical basis.

III. COMPONENT THERMAL ANALYSIS

The desire for high specific output with reduced heat rejection makes thermal loading of advanced in-cylinder components an important design parameter. With this in mind, various thermal analyses were conducted to guide the selection of design concepts.

The most basic analysis was a simple one-dimensional heat transfer study of a cast iron combustion chamber wall with and without plasma sprayed zirconia insulation. This geometry would be indicative of a cylinder head or liner.

Calculations were performed for water cooled, oil cooled, and uncooled configurations. Constant combustion gas side conditions were kept for all analyses for consistency, even though the wall temperature does affect the gas side boundary conditions in a real engine. Gas side boundary conditions were based on cycle simulation results. Coolant side boundary conditions were based on the literature and previous work at Cummins. The iron wall was considered to be .300 inch thick with a conductivity of 28 BTU/hr-ft-F. The Zirconia coating was .080 inch thick with an assumed conductivity of .7 BTU/hr-ft-F. The boundary conditions are summarized below.

Fluid	Temperature Degrees F	Heat Transfer Coefficient BTU/hr-ft**2-F
Combustion Gas	1400	158
Water	190	1000
Oil	230	500
Air	100	43 (uncooled)

A complete set of results is shown graphically in Figure 15. Several interesting observations can be made. If the primary goal is to reduce heat transfer, large changes can be made by either insulating the inner surface or by not cooling the outer surface. Once the inner surface is insulated, the cooling level on the outer surface does not have a large impact on the heat flux. This is shown in Figure 16. For durability of the base iron material though, the external cooling is very important. Figure 17 shows that the maximum metal temperature is somewhat determined by the presence of insulation, but more strongly controlled by the amount of cooling present. For durability of the ring pack and lubricant stability (in the case of a piston or liner) and combustion effects due to wall temperature, the maximum surface temperature is important. As shown in Figure 18, oil cooling will cause a moderate rise in surface temperature compared to water cooling, but a much greater change occurs if the wall is insulated or uncooled.

A more detailed thermal analysis was done of cylinder head, cylinder liner, and piston concepts using finite element techniques. To establish in-cylinder boundary conditions, cycle simulation was performed using TRANSENG. The model used an L10 engine rated at 350 HP @ 1800 rpm for 252 psi BMEP. These results along with an empirical database based on piston temperature measurements established the boundary conditions. A cycle average gas temperature was established at 1540 degrees F. The heat transfer coefficient varies radially from the center of the bore, having a maximum at the location of the outer edge of the piston bowl. The coefficient is constant circumferentially.

A. PISTON/LINER ANALYSIS

A finite element model with enhancements to reflect the moving piston to liner interface was used. This model was previously constructed at Cummins and utilizes slider-crank calculations to determine the varying thermal contact position between the liner and the piston/ring assembly. Multiple convective links are then made between the piston, piston rings, and liner to reflect varying contact positions. Gas side boundary conditions are cycle averages. The ANSYS finite element code is then used for the thermal calculations.

The model was first used to evaluate the effects of an insulated ring carrier in an aluminum piston. Since the temperature of the top piston ring and surrounding oil film are important criteria in determining an engine's maximum output capability, lowering their temperature would increase the BMEP potential. A thermal barrier between the hot combustion chamber area of the piston and the ring should lower the ring temperature. The conservative approach was taken in the model in that high convection was assumed between the piston and liner from the lands between the rings. This would reflect the case where the inter-ring

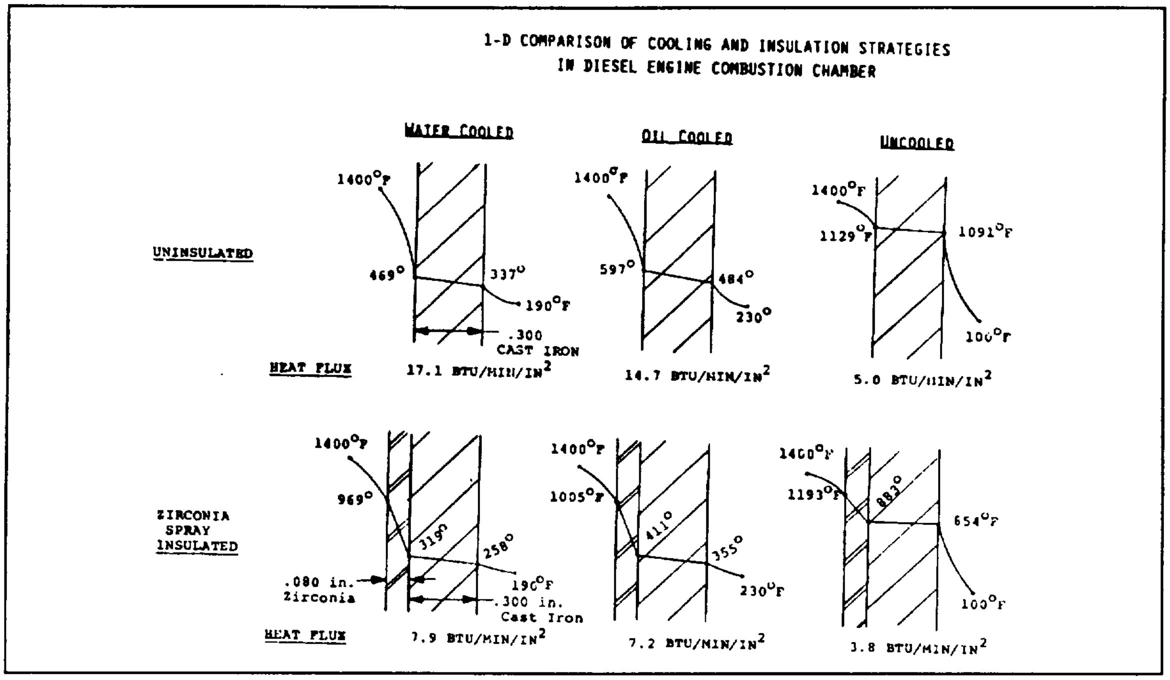


Figure 15

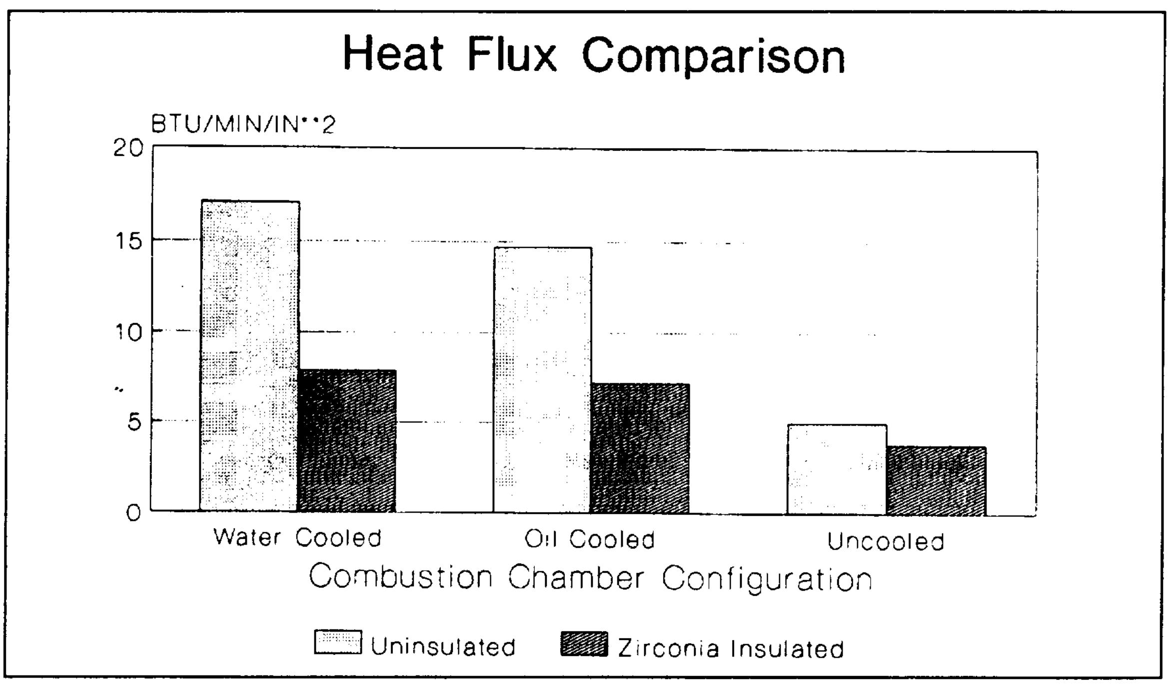


Figure 16

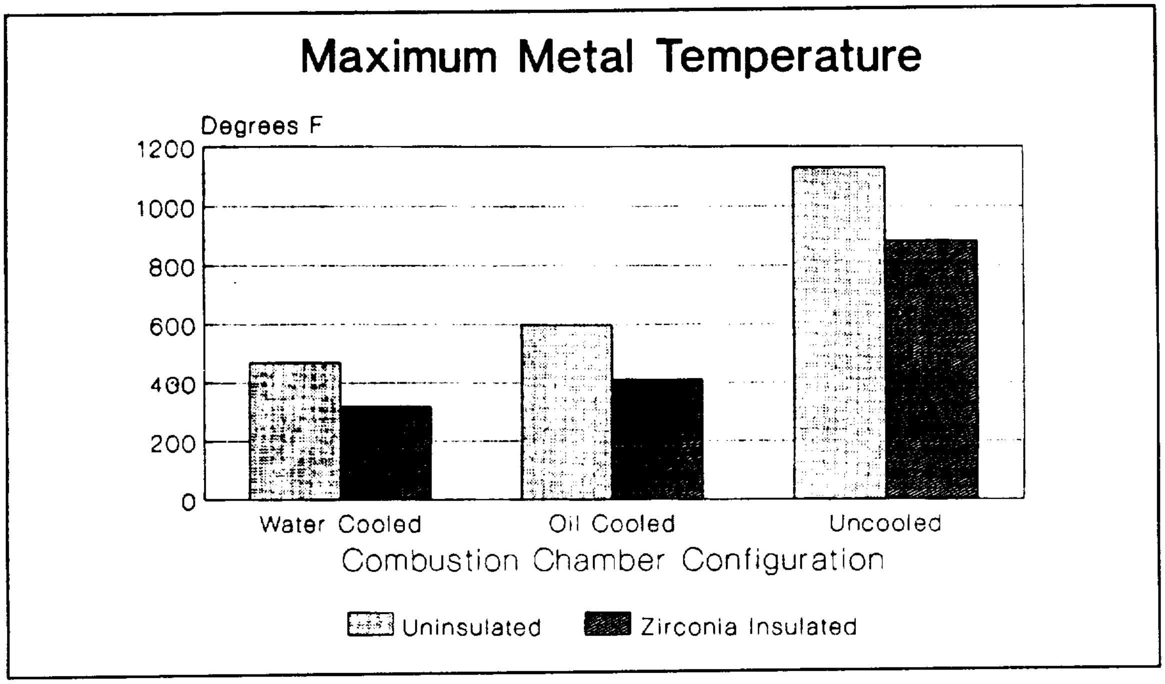


Figure 17

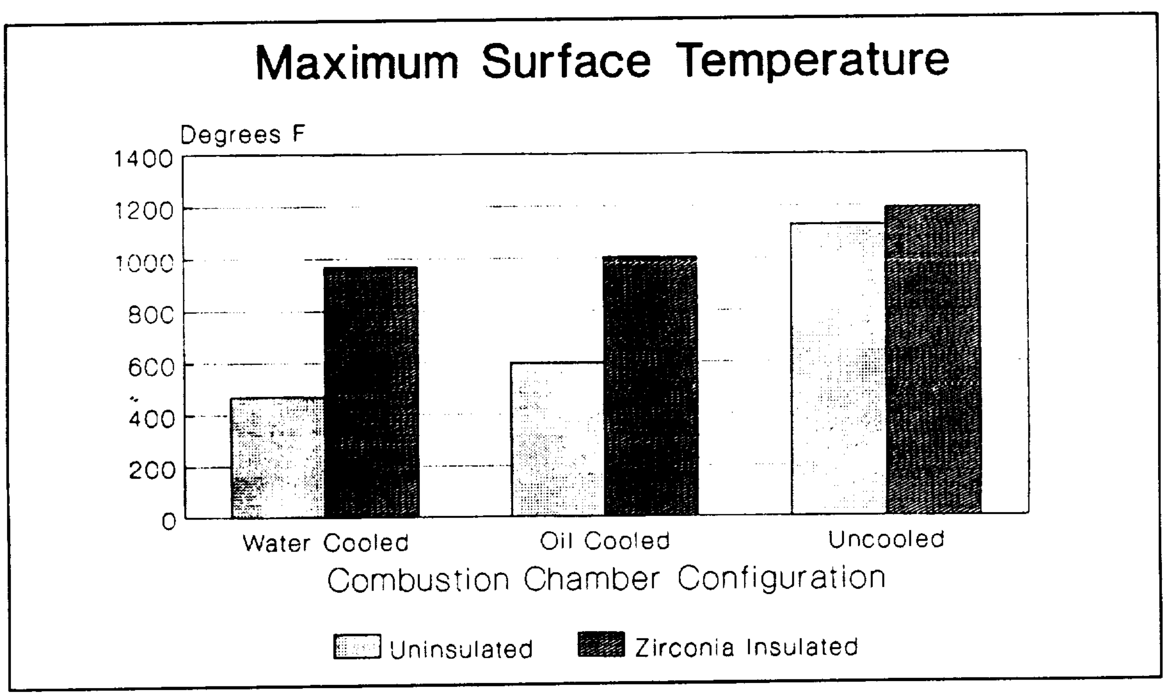
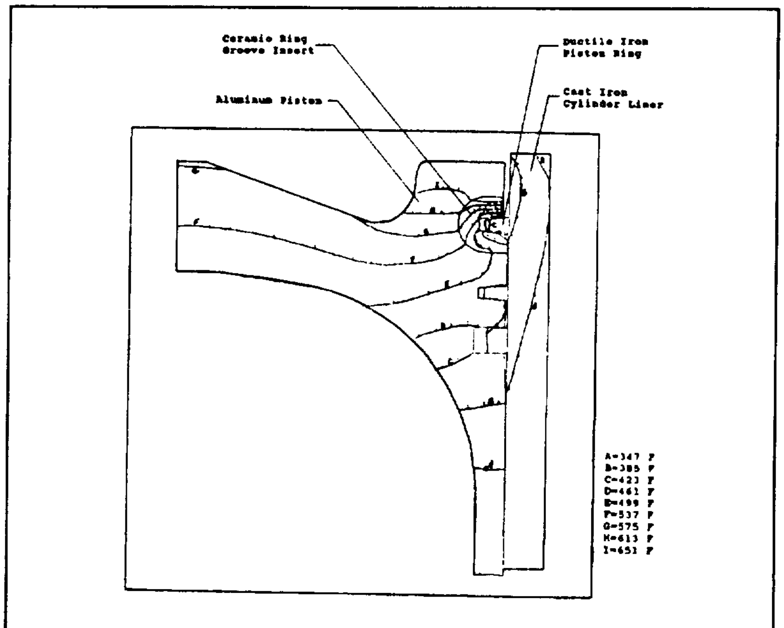


Figure 18

space is fully flooded with oil for best heat transfer, diminishing the importance of heat transfer through the rings. Even with this conservatism, an insulating ring groove material was shown to be an effective means of reducing top ring temperature. In should be noted that in a typical production aluminum piston, there is a Niresist insert cast into the aluminum for the top one or two rings. Niresist has a thermal conductivity that is approximately 20% of the base piston aluminum alloy conductivity so there is some ring groove insulation inherent in current aluminum piston designs. However, a ceramic ring groove insert would have thermal conductivity of approximately 2% of that of the base aluminum alloy for much more insulation effect.

The projected effect of a ceramic ring groove insert was predicted by the FE model. Figure 19 shows the predicted temperature profiles for the case where a ceramic groove insulator is in place in the top ring position. The inflection of the isotherm lines at the aluminum/ceramic interface as well as the large gradient across the ceramic show the insulation effect. The average top ring temperature with the ceramic groove is approximately 50 degrees F lower than with a Niresist ring carrier. Based on equivalent top ring temperature only, an increase in BMEP of around 20% would be allowable.



Piston Temperatures with Insulated Ring Insert

Figure 19

Piston design concepts using a ceramic ring groove have been discussed with the piston sub-contractor, Kolbenschmidt AG. Manufacturing processes to secure the ring into the piston are seen as the major hurdle, followed by ring wear due to the ceramic interface. One possible option is to use plasma sprayed ceramic on the back side of a metal ring carrier to avoid the potential wear issue. Casting in the insert securely would still be a major problem. The insulated ring carrier feature will not be a part of the primary piston concept at this time due to these issues, but will be considered at a lower level of effort.

CYLINDER LINER ANALYSIS

Starting with the conventional L10 water cooled cylinder liner, analysis was performed to determine the effects of cooling jacket changes on the piston and ring temperatures. The current cooling jacket is quite short by heavy duty diesel engine standards, approximately 3 inches in length. In the model the length was progressively reduced to determine the effects. Reducing the liner cooling jacket length had a minor effect on total heat rejection. The amount of heat transferred directly from the gas to the liner was slightly decreased while the heat transferred from the gas to the piston remained essentially constant. The heat transfer away from the piston was somewhat shifted from the cylinder liner to the oil on the underside of the piston. As seen in Figure 20, reducing the cooling jacket length has an adverse effect in the temperature of the top ring and piston crown. In light of the negligible effect on heat rejection, it is not recommended to reduce the length of the cooling jacket below 3 inches.

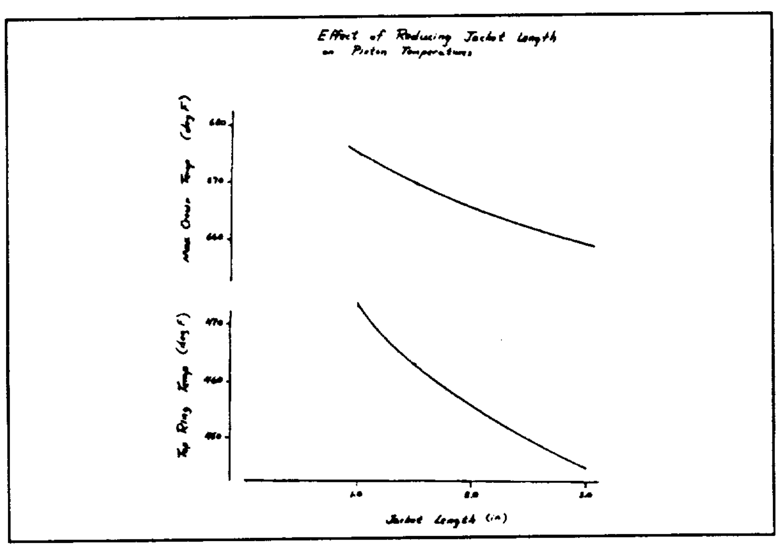


Figure 20

Changing the cooling fluid from a water-ethylene glycol mixture (simply referred to as water in most of this report) to an alternate coolant was also studied with the previously described model. While pure water is the most effective cooling medium known, the common and practical mixture of 50% water and 50% ethylene glycol(EG) is also effective because of the water content. Two commonly proposed alternate coolants are 100% propylene glycol or lubricating oil. The cooling property change from water/EG to propylene glycol is roughly 2/3 of the change from water/EG to lubricating oil and will be covered in more detail later in section C., COOLING SYSTEM ANALYSIS. In most heat transfer comparisons between water and oil, propylene glycol will yield results nearer to lubricating oil. In the model, the combined fluid and thermal effects of the coolant are expressed as a heat transfer coefficient. For a typical water cooled liner this value will be approximately 4 BTU/hr-in-F. Simply substituting oil into the water jacket will decrease this to about 1. The bulk coolant temperature will almost certainly go up with an alternate coolant. In the case of lubricating oil, it is beneficial to keep the temperature high to minimize hydrodynamic friction in the engine. In the case of either oil or a pure glycol, size limitations of the ambient heat exchanger (radiator) will cause coolant temperatures to rise. The combined effects of lower heat transfer coefficient and higher temperature combine to increase cylinder kit temperatures. Figure 21 plots this effect and points out the change in top ring temperature if current engines would have oil substituted for water. Not only does oil cooling cause ring temperatures to increase, the curves in this area are quite steep so that small changes to fluid conditions would lead to large temperature variations. The basic thermal properties of oil cannot be enhanced, but by changes to the coolant passages themselves, the heat transfer coefficient can be increased to approximately 3 BTU/hr-in-F with the resultant drop in ring temperature as shown. These enhancements include control of passage sizes to maintain high velocity and "turbulators" to both increase turbulence and add heat transfer surface area. It will be difficult for an oil cooled liner to cool as effectively as a water cooled liner, but with suitable design modifications, the negative impact can be minimized. The same holds true for propylene glycol. Simple substitution of propylene glycol for a water-ethylene glycol mixture will result in significantly higher temperatures in the piston, rings, and liner.

Considerable analysis and testing of insulated and/or uncooled cylinder liners has been performed at Cummins in the past. Two results are universally obtained: 1) volumetric efficiency decreases, and 2) piston, ring, and liner temperature increase. A secondary effect of increased temperatures is a decrease of the viscosity of the cylinder liner oil film and can result in lower friction. Heat rejection from the liner will decrease somewhat, however heat rejection from the piston to the lubricant will

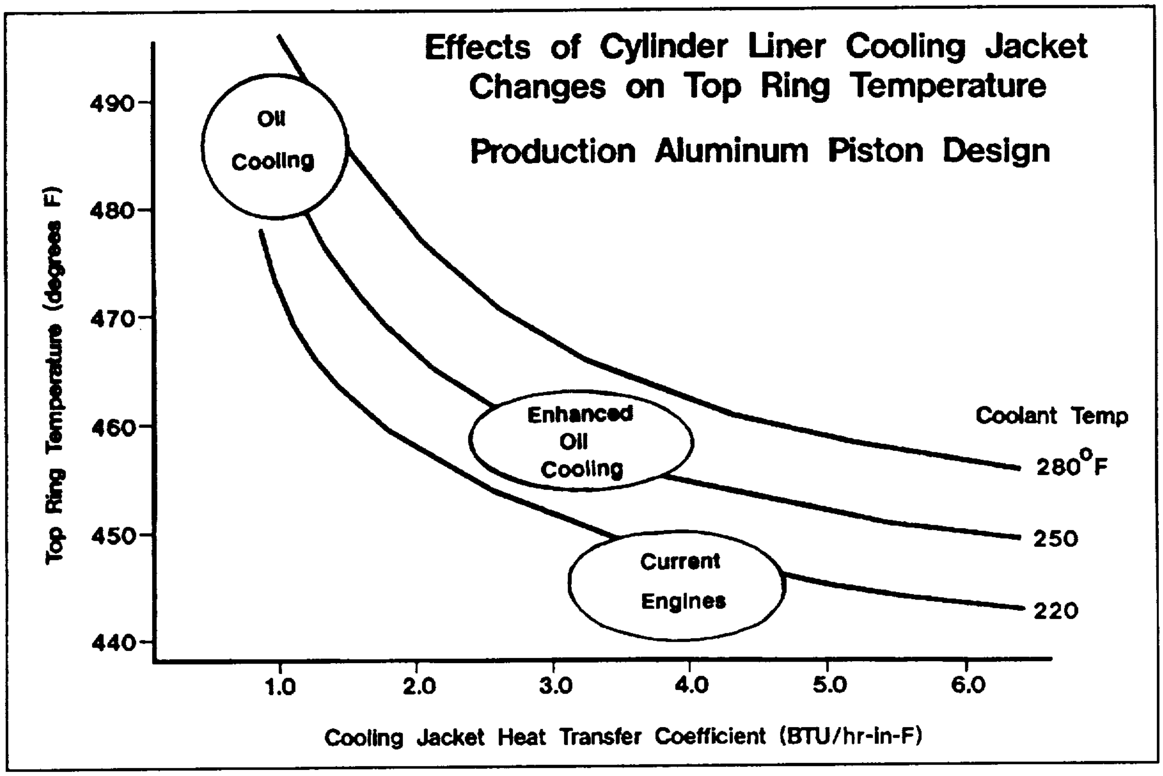
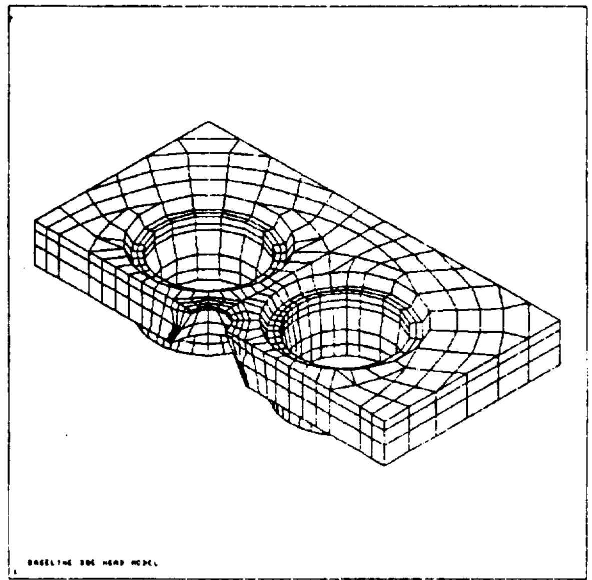


Figure 21

increase. The overall expected performance improvements are small, if existent at all. The mechanical durability problems of higher cylinder kit temperatures are definite and proven. It was concluded that insulated cylinder liners will not be pursued. A brief study was performed of the effects of insulation above the top ring reversal area of the liner only. With the proposed high top ring position, the benefits were minimal. As the final concept design progresses, however, insulation of the upper liner may be included if there are perceived performance benefits.

B. CYLINDER HEAD ANALYSIS

MODEL DESCRIPTIONS - Central to this analysis was the use of a three-dimensional finite element model of the cylinder head firedeck, as shown in Figure 22. The L10 utilizes a four valve cylinder head, with a centrally located injector, allowing a plane of symmetry to be assumed through the bore centerline, as shown in the figure. One intake and one exhaust valve were modeled, with the other valves assumed to be mirror images. Only the firedeck, and the first 25 mm of the port walls were modeled. The geometric, and material property details of the valve seats and the geometric details of the injector bore were included. Careful consideration was given to the thermal boundary



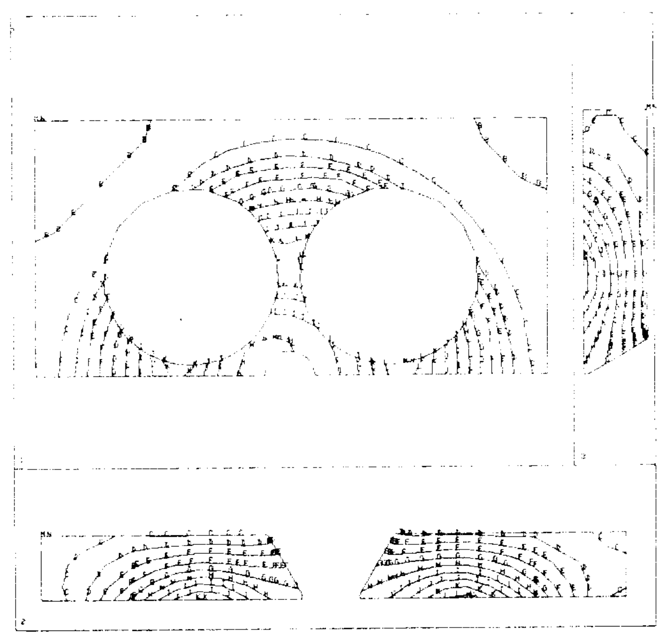
Cylinder Head Firedeck Model

Figure 22

conditions chosen to represent the combustion chamber. A time averaged gas temperature was supplied, through the use of TRANSENG, to be representative of the operating conditions studied. The combustion chamber heat transfer coefficient was varied in the radial direction, based on a substantial prior database of measured cylinder head temperatures. The heat transfer coefficient continually increased from the injector to the approximate radial position of the piston bowl edge, and then dropped rapidly to the perimeter of the cylinder. Previous use of this variable heat transfer coefficient was found to provide good agreement with measured temperatures, when used with a constant gas temperature. It is recognized that in the actual engine, both the temperature and heat transfer coefficient vary with space and time, and thus the heat transfer coefficient used here was an empirically determined function which lumped together the changes in gas temperature with position, the combination of radiation and convection heat transfer, and the changes with time and position of the convection heat transfer coefficient. Models such as the one used in this analysis are crucially dependent on accurate in-cylinder boundary conditions, but far less sensitive to the conditions applied to the rest of the model. This is due to the fact that in other regions of high heat transfer, such as convection to the cooling jackets, the convection coefficient is relatively high, thus resulting in a less dominant thermal resistance. In regions where the heat transfer coefficient is low, such as free convection from the outside of the engine to the ambient, heat transfer rates are low. Care was taken to choose thermal boundary conditions which would accurately represent convection to the coolant and ambient air, and conduction to other portions of the engine.

BASELINE CONDITIONS - For the purpose of this analysis (as well as all other Phase 1 component analysis) a rating of 260 KW (350 HP) at 1800 RPM was chosen. Each of the concepts to be assessed was compared at this operating condition. It is noted that in the development of the engine a full-load torque curve must also be considered. However, in this analysis, only the rated conditions were considered, recognizing that the conclusions drawn concerning the relative merits of the various concepts would be equally valid under peak torque conditions.

Heat transfer analysis was conducted, at the 260 KW rating, for the baseline engine. By definition, the baseline engine used all conventional, or current production, components. The resulting temperature profiles for the cylinder head firedeck are shown in Figure 23. The heat rejection for the entire combustion chamber is summarized in Table 1, where the heat transfer to each component is shown in the first column, and the heat load to the different cooling mediums is shown in the second column.



MX = 837
MN = 290

B = 300
C = 350
D = 400
E = 450
F = 500
G = 550
H = 600
I = 650
J = 700
K = 750
L = 800

Baseline Cylinder Head Temperatures

Figure 23

Table 1

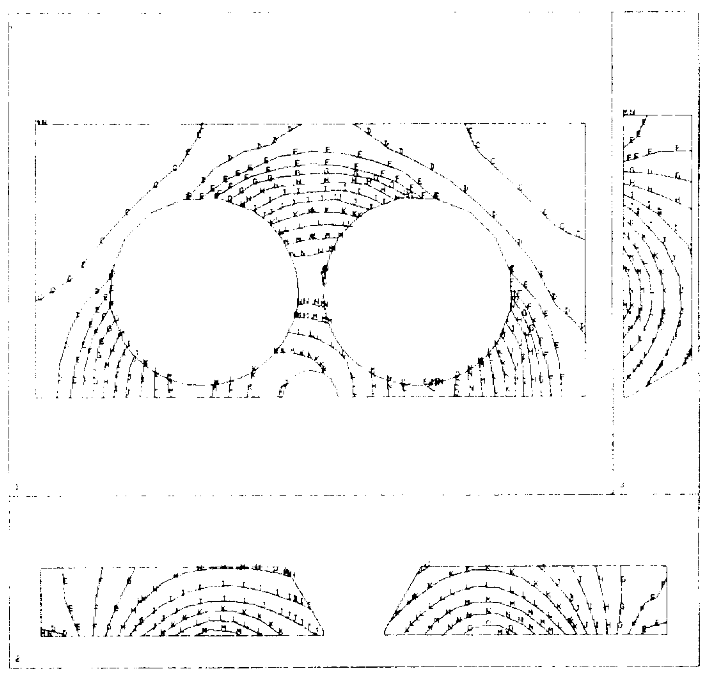
Baseline Heat Transfer Summary
260 KW @ 1800 RPM

	Heat Transfer (KW/Cylinder) From Gas	To Cooling Fluids
Gas to Piston	5.36	
Piston to Oil (remainder to liner)		3.59
Gas to Liner	4.61	
Liner to Coolant		5.33
Liner to Oil		1.05
		TOTAL
In-Cylinder Heat Rejection		13.52
Exhaust Port to Head	1.30	
Head to Coolant		4.85
		TOTAL
Heat Rejection to Coolant (not including friction)		14.82

The heat rejection through the liner includes that due to friction between the piston, rings, and liner. The total heat rejection from the engine would include the terms given here plus the additional friction from other areas of the engine, and that associated with the charge air cooler.

CYLINDER HEAD CONCEPT ANALYSIS - The cylinder head analysis goals involved the assessment of various design changes intended to reduce heat rejection, reduce engine cost and complexity, and control critical temperatures to provide acceptable thermal fatigue life. The analysis methodology involved the determination of the temperature field and heat rejection through the cylinder head firedeck, and the calculation of thermal stresses based on the temperature field. A Goodman diagram analysis was then used to calculate the fatigue life throughout the structure, based on the local temperatures and stresses, and cyclic loading between full power and idle conditions. The fatigue life reported is then the minimum life predicted in the structure, which in the case of this analysis, occurred adjacent to the injector bore, on the exhaust side of the firedeck. The use of a firedeck model to predict thermal stresses was previously demonstrated to be accurate, based on the recognition that, above the first water jacket, the majority of the cylinder head is at an approximately uniform temperature [1]. The use of a Goodman diagram analysis to perform cylinder head fatigue life calculations has also been previously reported [2].

The first case considered involved the replacement of the conventional water/ethylene glycol coolant with lubricating oil. Such a change would allow a significant simplification of the cooling system, and would be expected to result in a reduction in heat rejection through the cylinder head, due to the reduced convection heat transfer coefficient associated with the oil. The analysis was done by simply replacing the water with oil, while using the same cooling jackets. Boundary conditions in the cooling jackets were changed to represent the heat transfer coefficients expected with oil flowing at the same velocities as the water/glycol mixture. The resulting temperature profile is given in Figure 24, and the results referred to as "flooded oil cooling" are compared to the baseline in Table 2. Heat rejection through the cylinder head was reduced by 15 percent. However, the ineffectiveness of the lubricant as a coolant resulted in an increase in the maximum cylinder head temperature from 447 degrees C to 542 degrees C. The significant rise in temperature resulted in a 93 percent reduction in predicted thermal fatigue life.



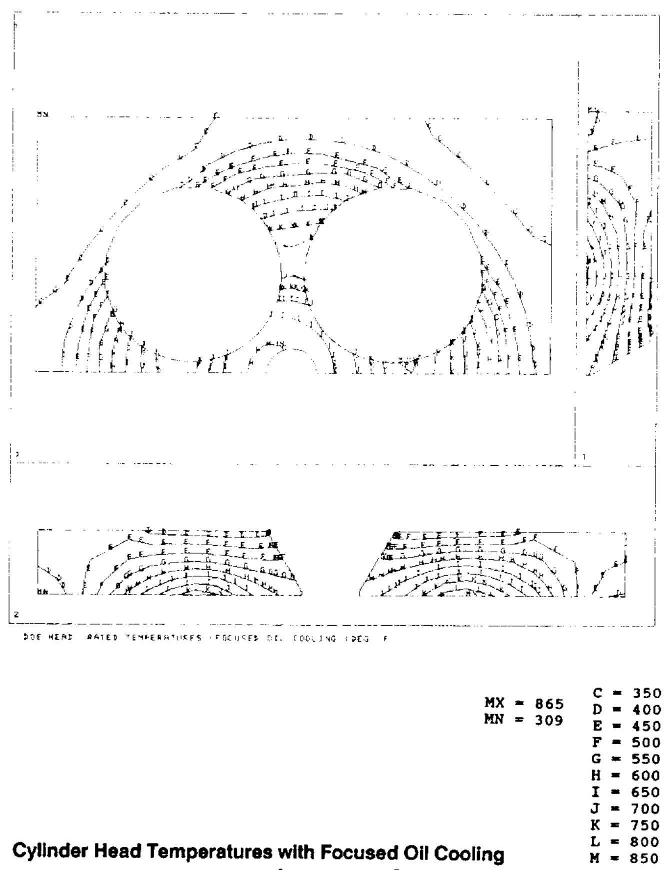
MX = 1008
MN = 311

C = 350
D = 400
E = 450
F = 500
G = 550
H = 600
I = 650
J = 700
K = 750
L = 800
M = 850
N = 900
O = 950
P = 1000

Cylinder Head Temperatures with Flooded Oil Cooling

Figure 24

Based on the preceding result, it was concluded that for oil to be effectively used as an engine coolant, steps must be taken to enhance its effectiveness in the regions where it is most needed. Heat transfer enhancement, through the use of rifling, or turbulators, was assumed in the valve bridges, and around the injector bore, where cooling is most needed. The oil was assumed not to flow through the regions around the bore perimeter, where it would not be needed. The temperature profile, for this case of "focused oil cooling" is given in Figure 25, and the results are again summarized in Table 2. For this case, the heat rejection was found to be reduced only 4.5 percent from the baseline, while the peak temperature was closer to the baseline, at 463 degrees C. Although the peak temperature was still higher than that of the baseline, the thermal fatigue life actually increased by 22 percent over that of the baseline. This may be explained by comparing the isotherm plot of the baseline condition, as given in Figure 23, with that given for the focused oil cooling, in Figure 25. Although the peak temperature is higher with the focused oil cooling, the temperature around the bore perimeter has also been increased, due to the reduction of cooling in this area. The now warmer perimeter region expands more than in the baseline case, thus providing less constraint to thermal expansion in the peak temperature regions, and reducing the compressive thermal stress in these regions.



Cylinder Head Temperatures with Focused Oil Cooling

Figure 25

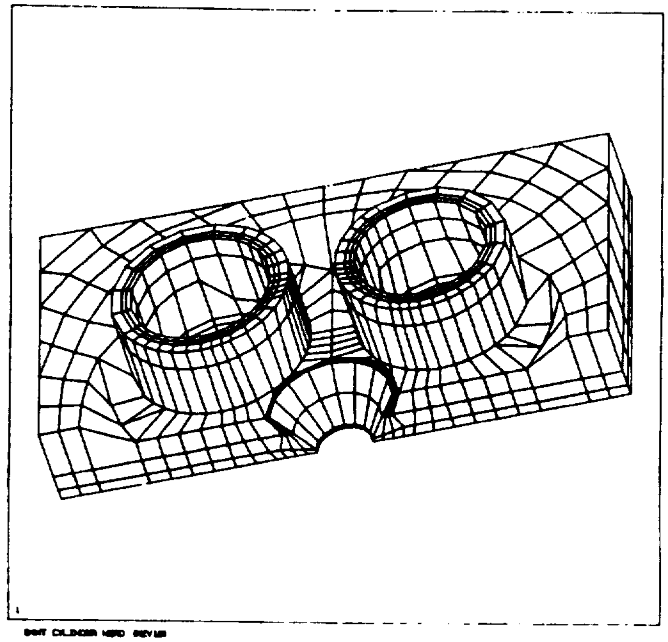
Table 2

Summary of Cylinder Head Analysis

	Heat Rejection KW/Cylinder	Max Temp (Deg.C)	Relative Fatigue Life
Baseline	3.52	447	1.0
Flooded Oil	2.99	542	0.07
Focused Oil	3.36	463	1.22
Insulated (focused oil)	2.44	389 (Metal) 771 (Ceramic)	infinite unknown

In conclusion, this analysis has suggested a durability benefit through the use of focused cooling in the valve bridge and injector regions, and the reduction of cooling in regions where it is not needed. However, it must be noted that this analysis was conducted using only a firedeck model. The high compressive stresses in the center of the combustion chamber are driven not only by the temperature gradient just discussed, but also by the overall stiffness of the head. If cooling jackets are removed from the perimeter regions of the head, the stiffness of the head would be changed due to the additional material which, in all likelihood, would replace these jackets.

As a cross-check on the results reported in the preceding analysis, a similar model was run, where the deck thickness was increased around the perimeter, in the regions where the cooling jackets were eliminated. The modified model is shown in Figure 26. The fatigue life calculated with focused cooling, and the increased deck thickness, was reduced 30 percent from that of the baseline. Several important points result from this analysis. First, it suggests that further design changes must be incorporated along with focused oil cooling, if its full benefits are to be realized. Steps must be taken not to increase the stiffness of the head around the perimeter of the bore, as this is an important variable in the fatigue life of the firedeck. Second, these results point to some weaknesses in the use of simple firedeck models for cylinder head analysis. Although the remainder of the head operates at a nearly uniform temperature, it does contribute significantly to the overall stiffness of the head. Stress results, and fatigue life calculations, made with firedeck models, must be seen as relative numbers at best. Even making relative comparisons of design changes must be done with care, as any changes affecting the stiffness elsewhere in the head, as was the case in the analysis just reported, will not be accurately reflected.



Modified Firedeck for Focused Cooling

Figure 26

As a final case, the use of ceramic insulation on the combustion chamber surface of the head was analyzed for the focused oil cooled head. A layered coating, as previously analyzed for application to a piston crown [3], was used. The layers consisted of a 0.13 mm bond coat, two 0.5 mm layers of mixtures of zirconia and CoCrAlY, and a 1.5 mm layer of zirconia. Previous efforts on NASA contract DEN3-331, "Thick Thermal Barrier Coatings for Diesel Components", have demonstrated this coating to have good potential for insulating combustion chamber surfaces. The results of this analysis are again summarized in Table 2.

The heat rejection through the cylinder head was reduced by 30 percent. The maximum metal temperature was reduced to 389 degrees C, resulting in a predicted infinite fatigue life of the base material. However, little can be said about the fatigue life of the ceramic at this point.

SUMMARY OF RESULTS - The foregoing analysis has pointed to several advanced combustion chamber component concepts, which are recommended for further development in the second phase of the program. The engine configuration may be summarized as follows:

Cylinder Head:

* focused oil cooling
* heat transfer enhancement in the cooling passages
* layered ceramic coating
* head design to avoid increased stiffness in regions where cooling jackets have been eliminated

Cylinder Liner:

* oil cooled
* heat transfer enhancement in the cooling jackets
* no reduction in jacket length

Piston:

* details to be developed by Karl Schmidt
* ceramic insulator around compression ring grooves

The configuration just outlined results in the engine heat rejection summarized in Table 3. The in-cylinder heat rejection is predicted to be reduced by 30 percent compared to the baseline case.

Table 3

Recommended Advanced Configuration
Heat Transfer Summary

	Heat Transfer (KW/Cylinder) From Gas	Heat Transfer (KW/Cylinder) To Cooling Fluids
Gas to Piston	2.67	
Piston to Oil (remainder to liner)		1.79
Gas to Liner	4.45	
Liner to Coolant		4.57
Liner to Oil		0.76
		TOTAL IN-CYLINDER
Heat Rejection	9.56	
Exhaust Port to Head		0.53
Head to Coolant		2.97
Heat Rejection to Coolant (not including friction)		10.09

Results of the piston, cylinder liner, and cylinder head analyses have also been published in an SAE paper [4].

C. COOLING SYSTEM ANALYSIS

The cooling system was investigated in two different studies. The first is a comparison of currently used ethylene glycol/water mixtures versus straight propylene glycol as a coolant for advanced diesel engines. Propylene glycol has been proposed as a new engine coolant to eliminate some of the problems associated with the water content of current coolants such as boiling, cavitation, and corrosion. Propylene glycol is also less toxic than ethylene glycol. The second study examines the viability of lubricating oil as a coolant. Using lubricating oil as a coolant has several potential advantages. Elimination of the water system simplifies the engine by removing the water pump, filter, and oil to water heat exchanger. The cylinder block and especially the head castings can be simplified by removal of the water jackets. The most expensive and troublesome feature of most cylinder head castings is the water jackets. Corrosion, boiling, and water pump seal failures are some of the problems that would disappear if oil were used as a coolant. Radiator, heater core, and coolant hose design would have to reflect the use of a different coolant.

1. PROPYLENE GLYCOL STUDY

A cooling system analysis was done to determine the effect of operating with 100 percent propylene glycol coolant (non-aqueous) instead of the more usual 50 percent aqueous solution of ethylene glycol. This change has been proposed as a near term possibility, especially by the chemical companies that produce propylene glycol and deserves a rigorous technical review. Since the possibility exists of near term application of this coolant, the following analysis assumes that currently available petroleum based lubricants would be used in the engine.

The analysis was done using an electrical circuit analogy of the cooling system, in addition to a cooling network program. A schematic of the cooling system analyzed is given in Figure 27. The heat flow rates and temperatures assumed for the analysis are given in the following table:

Table 4
Cooling System Model Assumptions

Freon Compressor Heat Load	600.00 10.6	BTU/min kW
Air Compressor Heat Load	50.0 0.9	BTU/min kW
Charge Air Cooler	3450.0 60.7	BTU/min kW
Turbo Compressor Work	3770.0 66.3	BTU/min kW
Intake Manifold Temp	105.0 40.6	Deg. F Deg. C
Ambient Temperature	77.0 25.0	Deg. F Deg. C
Radiator Heat Rejection	5306.0 93.3	BTU/min kW
Oil Cooler Heat Rejection	1832.0 32.2	BTU/min kW

Table 5
Cooling System Configuration Definitions

	1991 Techn'y	1994 Techn'y	
Radiator Frontal Area	1000.0 6452.0	1000.0 6452.0	sq. in. sq. cm.
Fan Blade Projected Width	2.9 73.7	3.5 88.9	in. mm.
Fan Blade Tip Clearance	0.75 19.1	0.25 6.4	in. mm.
Fan Diameter	32.0 81.3	32.0 81.3	in. cm.
Fan Speed	2390.0	2390.0	rpm
Vehicle Restriction Factor	10.0	15.0	
Number of Radiator Tube Rows	3.0	4.0	
Radiator Tube Style	Smooth	Turbulated	

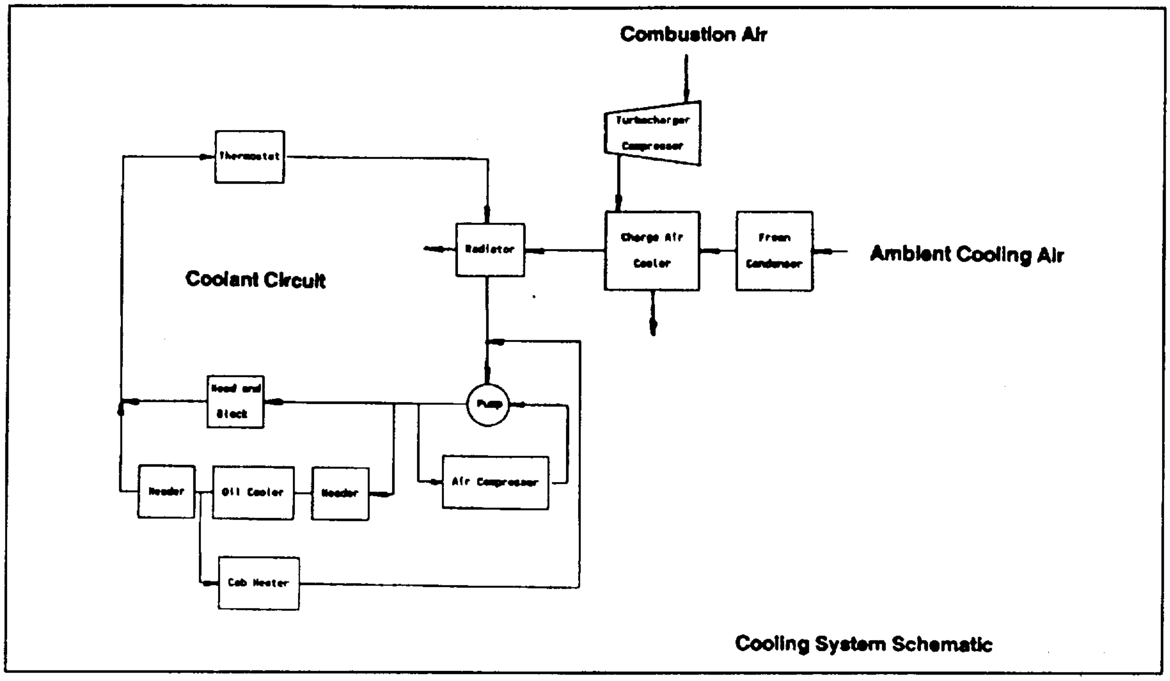


Figure 27

Three different configurations of the cooling system were considered for this study. These are summarized in Table 5. The configurations chosen to represent the 1991 and 1994 technology levels were taken from Cummins internal report, "X-Engine Cooling System Analysis" by A. R. Manon. It is expected that cooling fan technology will allow for less fan tip clearance, higher fan tip speeds, and larger blade projected widths, all of which increase fan air flow, and therefore cooling system performance in 1994. Conversely, it is likely that the air flow restriction under the hood will be increased due to the use of sloped truck hood designs which tend to lower the amount of air flow available for radiator cooling. In addition, the 1994 technology configuration takes into account more costly radiator designs which use coolant turbulators to increase the heat transfer between the fluids. The resulting maximum coolant temperature for each of the fluids is given in Figure 28. It can be seen that the fluid temperature is lower when ethylene glycol is the coolant than when propylene glycol is used. This can be explained by the lower thermal conductivity of the propylene glycol coolant. As Figure 28 also shows, the maximum temperature of the propylene glycol coolant can be lowered with 1994 technology, but in all cases the ethylene glycol also benefits from the changes, although not to the same extent. With the optimal 1994 technology configuration, the maximum propylene glycol temperature is about the same as that seen by the ethylene glycol solution in 1991.

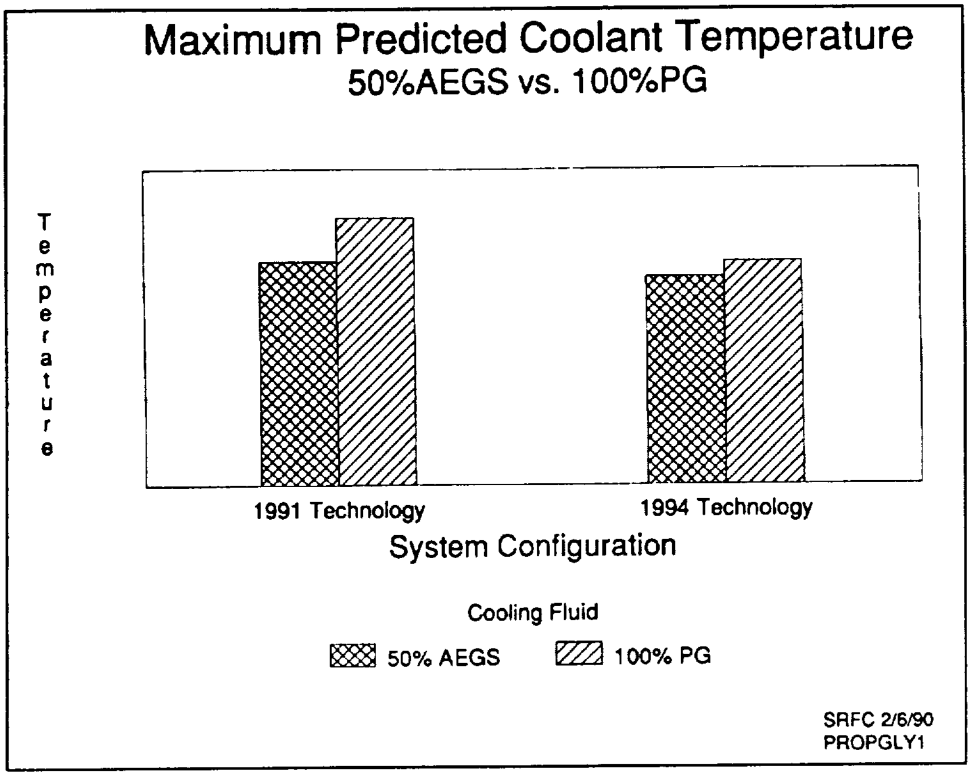


Figure 28

Figure 29 shows the maximum lube oil temperatures for the ethylene and propylene glycol cooled engines. As noted with the coolant temperatures, the lube oil temperature is also lower when ethylene glycol is used as the coolant, with less difference evident when 1994 technology is examined.

The increase in the maximum temperature of the coolant and the lube oil when the engine coolant is propylene glycol is quantified in Figure 30. The temperature of each fluid, when 50% AEGS coolant is used in a vehicle using 1991 technology, is taken as the baseline.

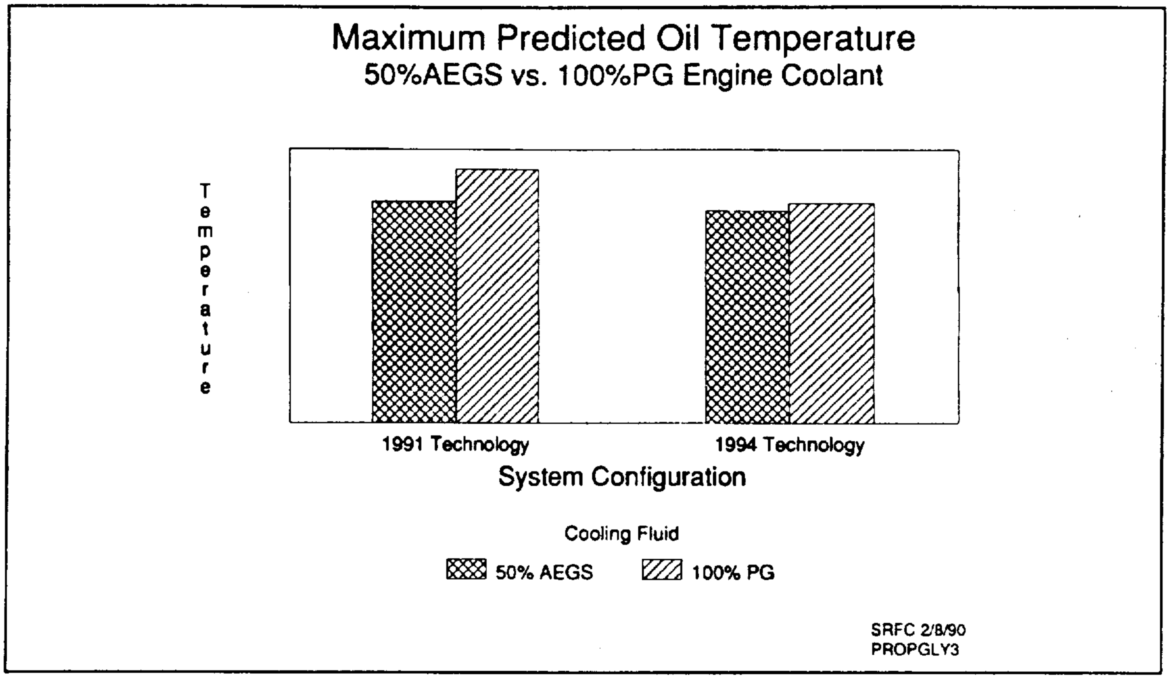


Figure 29

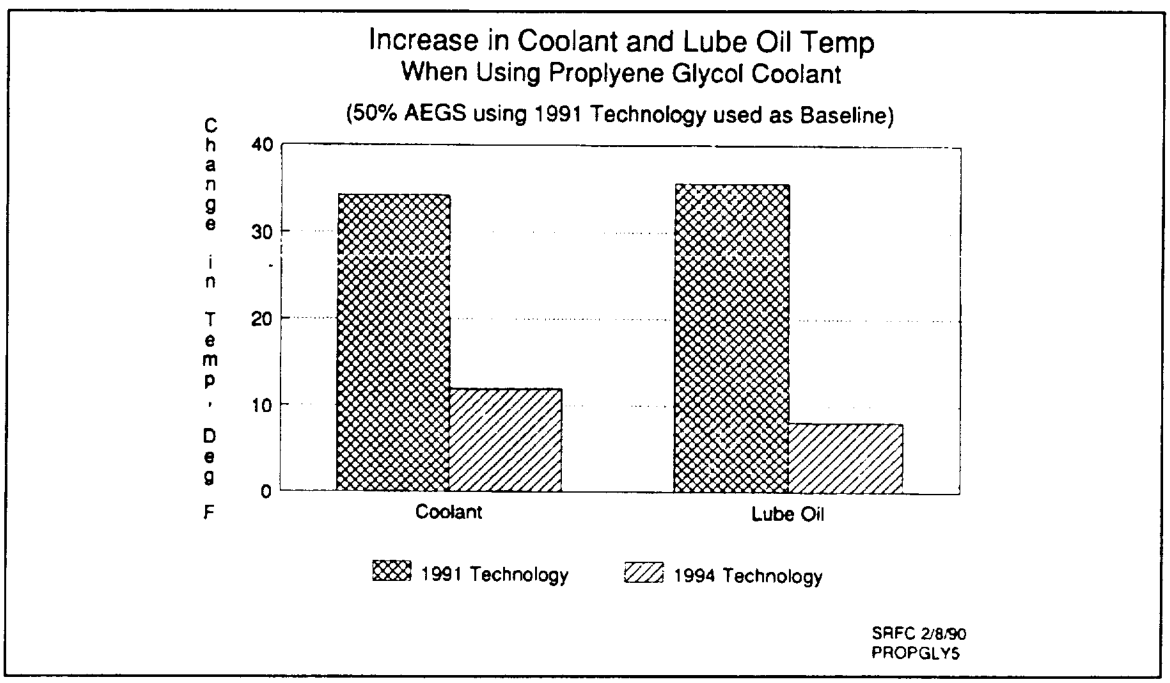


Figure 30

Oil degradation, deposit formation, and oil consumption are all exponential functions of temperature according to Cummins internal technical memo "Piston Strategy Concerns for '91 Engines" by K. L. Hoag. In addition, it is primarily the breakdown of the lube oil that contributes to such engine durability issues as high oil consumption, and blowby due to carbon buildup in the piston ring grooves [5].

From work reported by Hoag in Cummins internal report "Analysis of the Effect of Liner Jacket Design on Piston and Ring Temperatures of the 91 NT Engine", it can be concluded that a 30 deg. F. (16.7 deg. C.) increase in coolant temperature will increase the top ring temperature about 15 deg. F. (8.3 deg. C.) McGeehan, et. al., found that every increase in piston ring groove temperature of 54 deg. F. (30 deg. C.) doubled the amount of top ring groove deposits that were formed [5]. In addition, McGeehan also found that the bulk oxidation of the oil doubles for each 120 deg. F. (67 deg. C.) increase in top ring groove temperature. Therefore, the coolant and lubricant temperature increases predicted will lead to a decrease in engine durability. The baseline temperatures of the coolant and lubricant as shown in Figure 30 are determined to provide the engine with excellent durability. With the emphasis on longer-lived engines, it becomes critical to maintain fluid temperatures at current levels to avoid reducing engine life.

From these findings, it can be concluded that the use of 100 percent propylene glycol coolant as a substitute for a 50 percent aqueous solution of ethylene glycol will increase coolant and lubricant temperatures. It can, therefore, be assumed with high confidence that the durability of the engine will be lowered if propylene glycol is substituted for ethylene glycol.

2. USE OF LUBRICATING OIL AS A COOLANT

Thermal analysis of the critical engine components resulted in a recommended engine configuration that used lubricating oil as the engine coolant [4]. To determine whether lubrication oil would be a viable cooling fluid from a cooling system standpoint, cooling system models were built to predict the minimum radiator frontal area that would be required to control the maximum oil temperature.

Figure 31 shows a schematic of the oil-cooled engine cooling system. The lubrication oil flow and heat load is assumed to be the same as for a 1991 L10 engine. The flow in the head and block have been adjusted so that the oil temperature increase across each will be approximately the same. An air-to-oil heat exchanger is used to cool the oil in place of the radiator. The charge air system, air compressor, cab heater and thermostat are modeled as the baseline system.

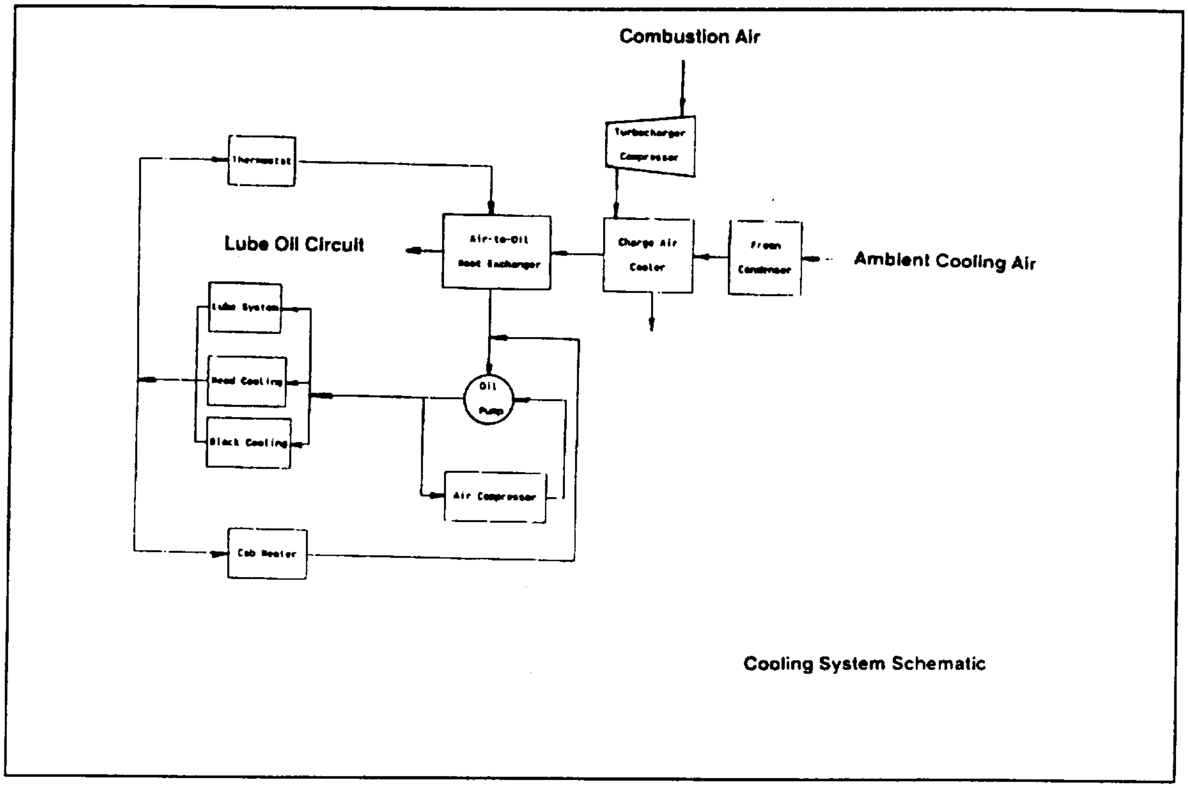


Figure 31

Table 6 summarizes the heat load and temperature assumptions made in the model. Included in the table for comparison purposes are assumptions used for a similar analysis using a 50 percent aqueous ethylene glycol solution (AEGS). As reported by Manon, there is a 5 percent reduction in the heat transfer through the cylinder head for the case using focused oil cooling when compared to flooded AEGS cooling. If a ceramic thermal barrier is applied to the head, the heat transfer is reduced 30 percent from the baseline cylinder head [4]. This is the case shown in Table 6. Due to the decreased thermal conductivity of the lube oil, the head and block will need to be redesigned to increase the cooling passage heat transfer coefficient. The flow restriction through the head and the block has been increased to take into account the greater pressure drop that is expected when heat transfer enhancement devices are used.

Table 6

Cooling System Model Assumptions

	Coolant		
	H_2O/EG	15W40 Oil	
Freon Compressor Heat Load	600.0 10.6	600.0 10.6	BTU/min kW
Air Compressor Heat Load	50.0 0.9	50.0 0.9	BTU/min kW
Charge Air Cooler	3450.0 60.7	3450.0 60.7	BTU/min kW
Turbo Compressor Work	3770.0 66.3	3770.0 66.3	BTU/min kW
Intake Manifold Temp	105.0 40.6	105.0 40.6	Deg. F Deg. C
Ambient Temperature	77.0 25.0	77.0 25.0	Deg. F Deg. C
Radiator Heat Rejection	5306.0 93.3	3692.0 64.9	BTU/min kW
Oil Cooler Heat Rejection	1832.0 32.2	(See Rad)	BTU/min kW

In order to determine the minimum frontal area that would be required with lube oil cooling, a factorial test was developed that evaluated the effects of various system parameters on the cooling performance of the engine. From this analysis, a regression curve was developed for each cooling system performance measure. These regression curves were then optimized for lowest radiator frontal area and lowest input power required. Table 7 summarizes the cooling system parameters that were varied. The configurations chosen to represent the 1991 and 1994 technology levels were taken from the previously mentioned Manon internal report. It is expected that cooling fan technology will allow for less fan tip clearance, and larger blade projected widths, both of which increase fan air flow, and therefore cooling system performance in 1994. Conversely, it is likely that the air flow restriction under the hood will be increased due to the use of sloped truck hood designs which tend to lower the amount of air flow available for radiator cooling.

Table 7

Cooling System Configuration Definitions

	1991 Techn'y	1994 Techn'y	
Fan Blade Projected Width	2.9	3.5	in.
	73.7	88.9	mm.
Fan Blade Tip Clearance	0.75	0.25	in.
	19.1	6.4	mm.
Vehicle Restriction Factor	10.0	15.0	

The minimum radiator frontal area required to maintain a maximum oil temperature of 250 deg. F (121 deg. C) with 1991 technology is predicted to be 699 square inches (4510 square centimeters). With 1994 technology, the minimum radiator frontal area required is 435 sq in (2810 sq cm). From these values, it can be seen that oil cooling of the engine is a viable option.

To determine the power input required for the different cooling systems, the horsepower required for the radiator cooling fan, and the oil pump were calculated. Figure 32 shows the power required for the 1991 and 1994 technology cases. If the total required horsepower is minimized, the radiator frontal required increases to 864 sq in (5570 sq cm); also shown in Figure 32.

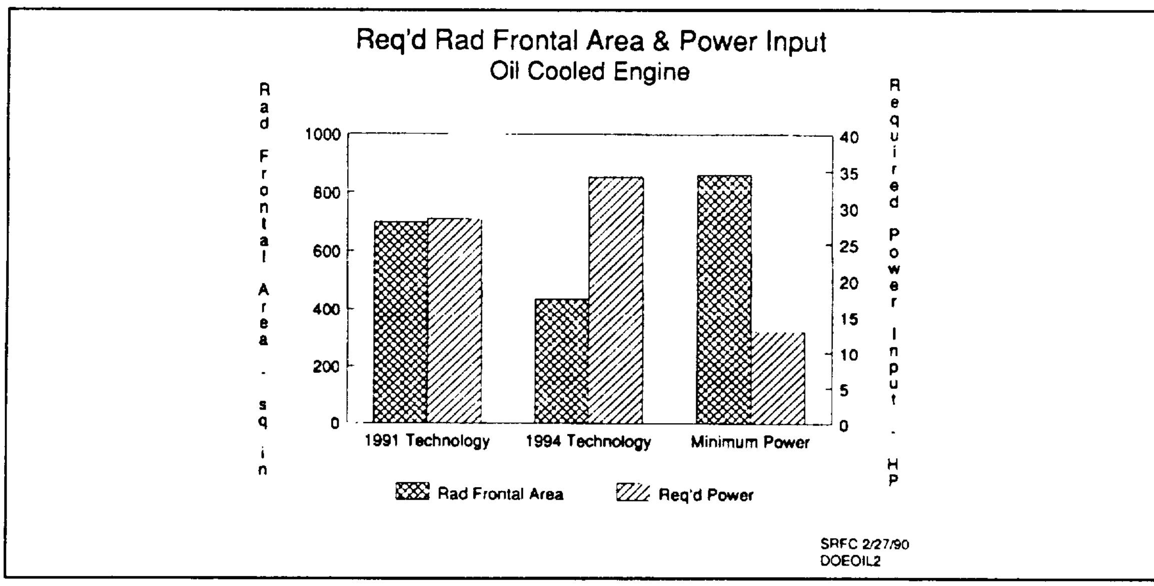


Figure 32

Figure 32 shows the tradeoff that is needed between minimizing the radiator frontal area, and minimizing the amount of power which is required to drive the cooling fan and the oil pump. The smaller radiator needs additional fan power to increase the air flow to keep the engine cool. On the other hand, if the radiator is allowed to increase in size, the amount of power needed to drive the fan to cool the oil is greatly reduced.

Conclusions

- Radiator frontal areas are not excessive when used with an oil cooled engine that has low heat rejection.

- There is a tradeoff between the power that is required to operate the radiator cooling fan and the oil pump, and the radiator frontal area.

IV. PISTON DESIGN AND ANALYSIS

In a heavy duty diesel engine the piston is probably the most critical component outside of the fuel system. The piston must form the combustion chamber, withstand severe mechanical and thermal loads, and prevent lubricating oil consumption. These functions must not degrade significantly with time due to wear, material failure, or deposit buildup. At the high output ratings foreseen for the year 2000, these requirements become even more difficult. Since piston design, analysis, and manufacturing is a specialized field, Cummins has retained the services of Kolbenschmidt AG (Karl Schmidt Corporation) to perform many of the piston related tasks in this program. The conceptual work has been a shared responsibility between Cummins and KS while the details of design and analysis has been performed by Kolbenschmidt. All major decisions have been mutually agreed upon before proceeding.

The initial task to be performed was to narrow the scope of possible alternatives. The approach that was taken is the following:

1. Identify design criteria and objectives.
2. Propose three likely concepts.
3. Evaluate the capabilities of the three concepts with finite element analysis.
4. Select the most promising concept.

The general design criteria is for the piston to be consistent with the overall concept of a low fuel consumption, low emissions engine of the year 2000. Specifically, this requires a piston capable of high thermal loads, preferably with low heat rejection. Initial screening of concepts was done assuming a 350 horsepower rating but uprate capability is required. Oil consumption must be reduced from current levels. Flexibility in the design of the combustion chamber must be maintained as low emissions combustion systems are developed. Reciprocating weight should be kept close to current levels. Any increased cost must be offset by increased output potential. The design should include features that will enable the development of advanced technology that will have broad application to future piston designs.

Three concepts were chosen that met the requirements of significant advances but appear feasible in a 10 year time span. These were then compared using finite element analysis to help in the selection of one concept for development in Phase 2. Combustion chamber thermal boundary conditions were the same as used in previously reported piston, liner, and cylinder head analysis. The three design candidates were:

1. Steel crown articulated piston with and without a thermal barrier coating.
2. Fiber reinforced aluminum piston with ceramic bowl insert and thermal barrier coating.
3. Spherical connecting rod/piston joint.

In the process of choosing three suitable concepts, numerous ideas were generated and weighed against the objectives. The following are some of the concepts that were discussed but not selected. An iron crown articulated piston has higher BMEP capabilities than an aluminum piston. The improvement was seen to be limited, and the technology essentially exists, so this concept was not considered further. A fabricated steel articulated piston has some interesting possibilities for low weight and high output. Costly welding and inflexibility in bowl design led to this concept being dropped. A one-piece ductile iron piston would be capable of high outputs. High cost and limitations to bowl geometry caused this to be eliminated. The most radical concept discussed was the use of an air bearing piston. This design would not use piston rings and not be liquid lubricated. The low friction and elimination of the piston as a source of oil consumption are very attractive. Investigation showed that clearance between the piston and cylinder wall would need to be on the order of .0005 inches (.013 mm). Manufacturing tolerances, thermal growth, and mechanical distortion make this seem like an almost insurmountable hurdle. The side thrust capability under even perfect conditions is suspect. Lastly, an entirely new engine configuration that keeps liquid lubricant away from the piston (or eliminates it altogether) would be required. With these arguments, it seemed obvious that air bearing pistons are further than 10 years away for heavy duty diesels, and any future practicality is in doubt.

STEEL CROWN ARTICULATED PISTON

The basic design features of this concept include a crown of high strength cast steel (similar to AISI 4140) and an aluminum skirt. Some unique features were included in an attempt to minimize heat rejection. No spray cooling with lubricating oil is in the design. The skirt contains a shield to minimize the amount of crankcase oil splash that contacts the underside of the crown. To compensate for the lack of cooling, the crown section between the bowl and the ring grooves is quite thick compared to other articulated piston designs. The design was analyzed both with and without a 2.5 mm thick plasma sprayed zirconia insulator on the combustion face. In the insulated case, the outside dimensions of the piston were kept constant and additional material was added to the undercrown to maintain section modulus. The insulating coating was the 3 layer coating developed under the "Thick Thermal Barrier Coatings" contract. Figure 33 shows a quarter section of the solid model of the insulated version of the piston. The oil splash shield is seen with openings for the crown support struts. The design has planes of symmetry along the two exposed faces. From the solid model a 3-dimensional finite element model was constructed.

ADVANCED IN-CYLINDER COMPONENTS PROGRAM

Articulated Piston with Zirconia Insulation

Solid Model

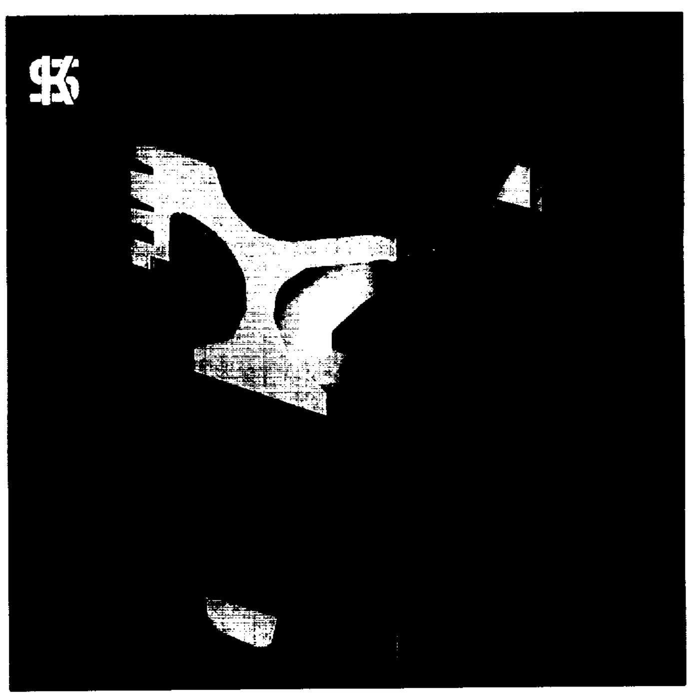


Figure 33

Figure 34 shows the predicted isotherms of the uninsulated version. Only the crown portion and the piston pin are shown. The peak temperature occurs in the center of the bowl and is 526 degrees C (979 F). The undercrown maximum is 505 C (941 F). The midpoint of the rear face of the top ring groove is 378 C (712 F). A detailed analysis was performed with both thermal loading and 3000 psi cylinder pressure. Based on the extensive Kolbenschmidt material database, fatigue failure of this design is predicted due to thermal and mechanical loads. The failure would originate in the bowl center, which is reported to be a common failure location for large bore pistons with ferrous crowns. Surfaces in contact with the lubricant are much higher in temperature than can be tolerated by any currently available lubricants. An advanced high temperature synthetic lubricant would be necessary for this design.

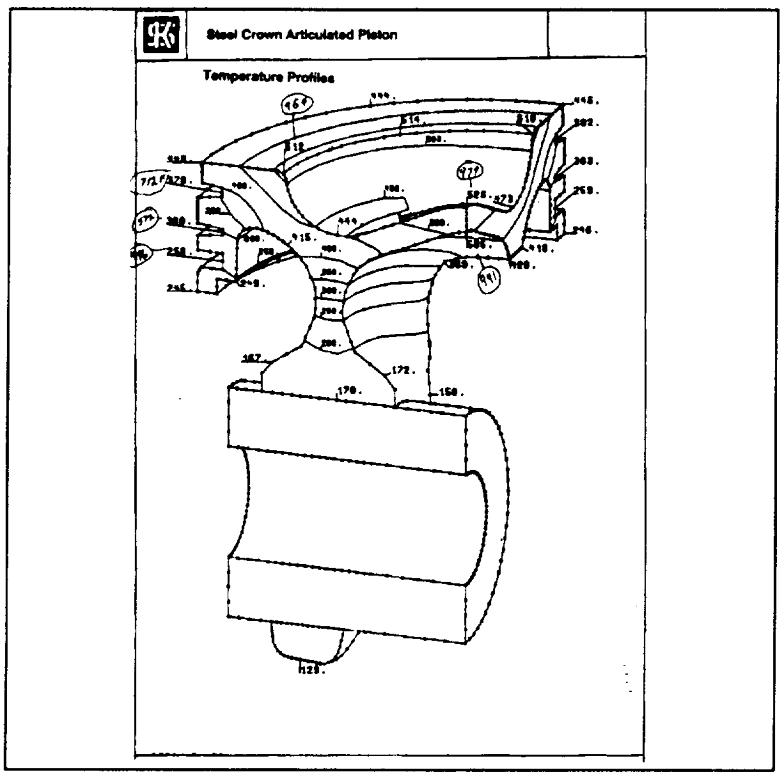


Figure 34

The design was then revised to include a 2.5 mm (.100 inch) thick thermal barrier coating of plasma sprayed zirconia. As previously mentioned, the crown design was modified such that the outer dimensions were the same with the coating as in the uncoated case. The undercrown area was modified to add material to compensate for the 2.5 mm of steel that was removed to make room for the insulating layer. As expected, the combustion surface temperatures increased and the metal temperatures decreased with the application of the thermal barrier coating. Figure 35 shows the predicted isotherms. Peak surface temperature is now 744 C (1371 F) at the bowl rim. Peak metal temperature is 380 C (716 F) in the bowl center. Peak undercrown temperature is 363 C (685 F) in the bowl center. The top ring groove is 300 C (572 F). All temperatures fall within the normal capabilities of the chosen materials. A problem of oil coking from the oil mist in contact with the undercrown would be expected with traditional lubricants.

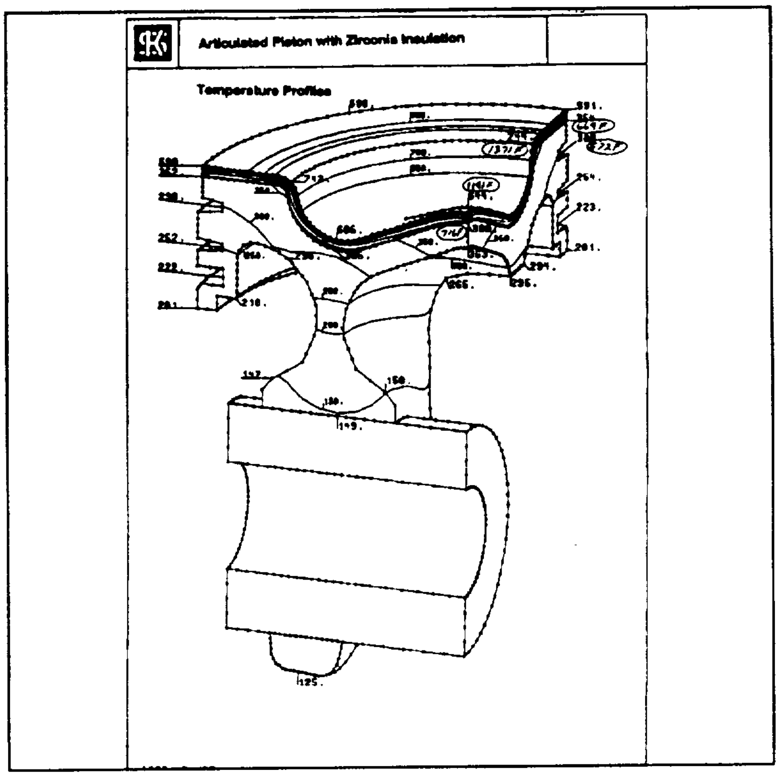


Figure 35

All pistons that use a pin to attach to the connecting rod demonstrate some amount of bending around the pin when subjected to high cylinder pressure. In one-piece aluminum piston designs, a tall compression height (distance from pin centerline to top of piston) and/or long skirt minimize this effect. In an articulated design, the crown is not attached to much of this supporting structure and relies on high material strength to resist bending. In Figure 36 the distortion under cylinder pressure loading is shown magnified 50 times. Along the pin axis, the piston crown compresses .120 mm (.005 inch). Perpendicular to the pin, the compression is .274 mm (.011 inch), or .154 mm (.006 inch) more than along the pin axis.

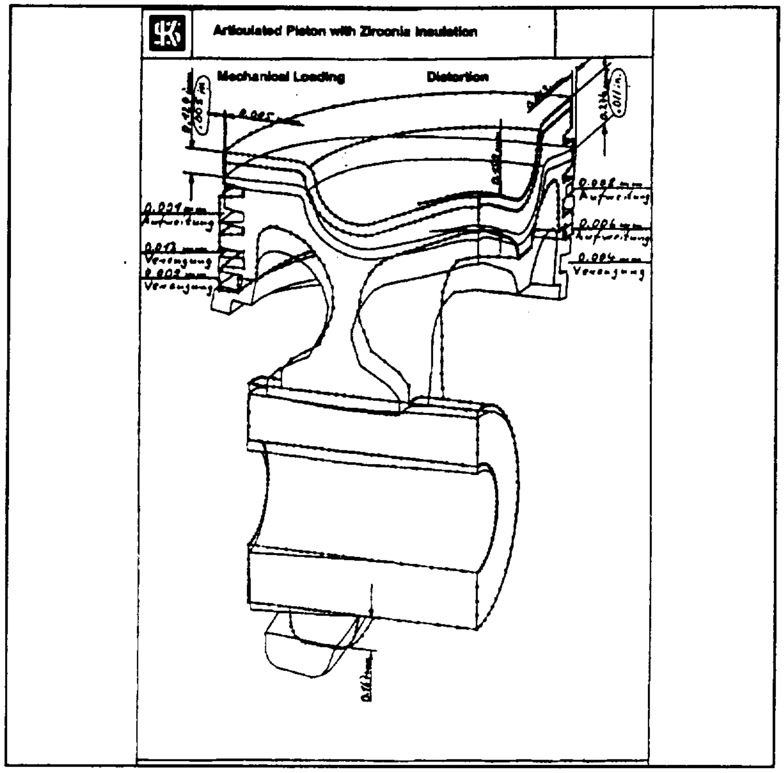


Figure 36

This bending causes distortion of the ring grooves which can lead to increased blowby and oil consumption. The main problem is the high stress at the bowl rim. Compressive stress occurs perpendicular to the pin and tensile stress occurs parallel to the pin. The maximum principle stress from mechanical loading at the predicted temperatures is shown in Figure 37. Stress from bending around the pin can be seen in both the thermal barrier coating and the underlying steel. Parallel to the pin there is a tensile stress of 248 MPa (36 KSI) in the steel and 38 MPa (5.5 KSI) in the coating. Perpendicular to the pin there is a compressive stress of 59 MPa (8.6 KSI) in the steel and 24 MPa (3.5 KSI) in the coating. These stress levels are acceptable for the steel material chosen. Coating durability is unknown, however the stress levels are below the measured strengths of the coating.

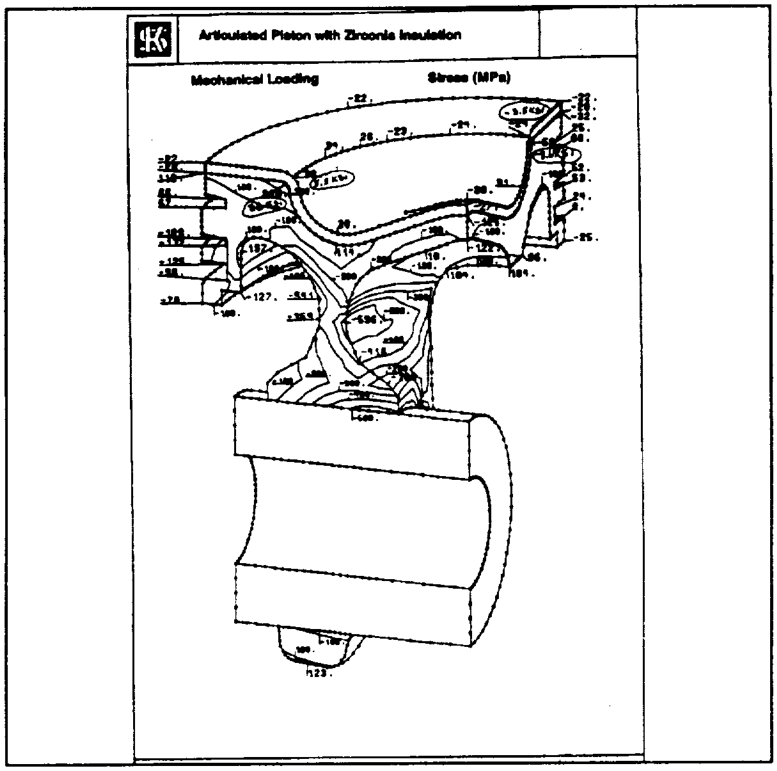


Figure 37

An uncooled steel crown articulated piston does not appear to be capable of meeting the program objectives. Application of a ceramic thermal barrier coating to the combustion surface would bring metal temperatures and stresses into an acceptable range and make the concept feasible for this program.

To further define the capabilities of an articulated piston, an analysis of the design will be performed in Phase 2 of the In-Cylinder Components Program with material properties of nickel aluminide (Ni_3Al) used for the crown. Nickel aluminide is an intermetallic material possessing very high strength at elevated temperatures. Cummins has a license from Oak Ridge National Laboratories to use nickel aluminide for diesel engine components.

INSULATED FIBER REINFORCED ALUMINUM PISTON WITH ALUMINUM TITANATE INSERT

A design concept was prepared that attempted to meet the program goals with an advanced aluminum piston. The base material was assumed to be squeeze cast aluminum alloy. Ceramic fiber reinforcement was inserted in the bowl area for improved fatigue strength. A cast in aluminum titanate insulator was placed in the combustion bowl and plasma sprayed zirconia applied to the bowl center and rim. The bowl shape was the same as used for the previous articulated piston analysis. First and second ring grooves are in a niresist insert. The model assumed there was no spray cooling of the underside of the piston, only crankcase oil mist. Figure 38 shows a quarter section FE model of this design.

ADVANCED IN-CYLINDER COMPONENTS PROGRAM

Insulated Fiber-Aluminum Piston

ORIGINAL PAGE
COLOR PHOTOGRAPH

Finite Element Model

Figure 38

Predicted temperatures are shown in Figure 39. The maximum temperature occurs in the cast in bowl insulator and is 639 C (1182 F). Maximum metal temperature occurs at the bowl edge in the fiber reinforced zone and is 378 C (712 F). The predicted temperature levels are generally within acceptable limits for the specific materials. The combination of thermal gradients and different thermal expansion rates within the different parts of the structure has led to a great deal of thermal distortion. A plot of the distorted shape magnified 100x is shown in Figure 40.

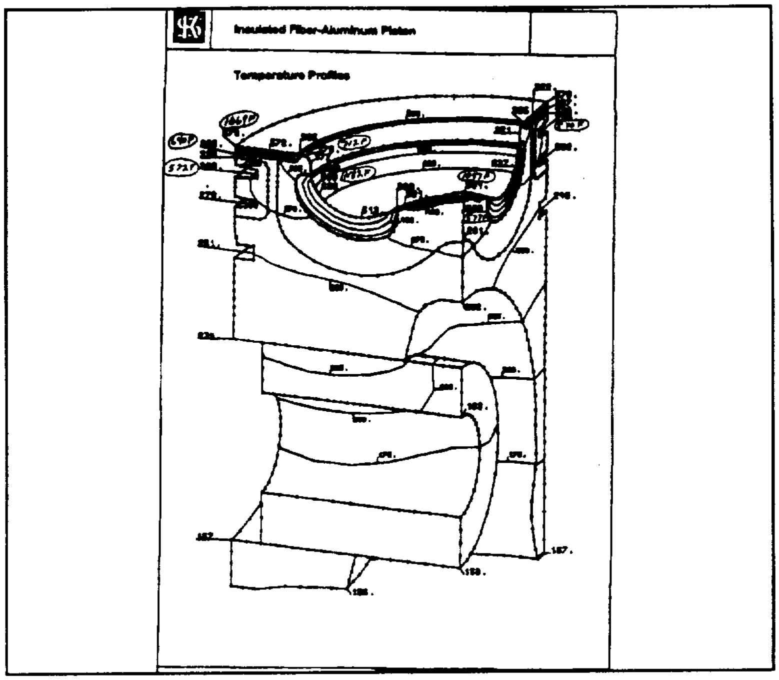


Figure 39

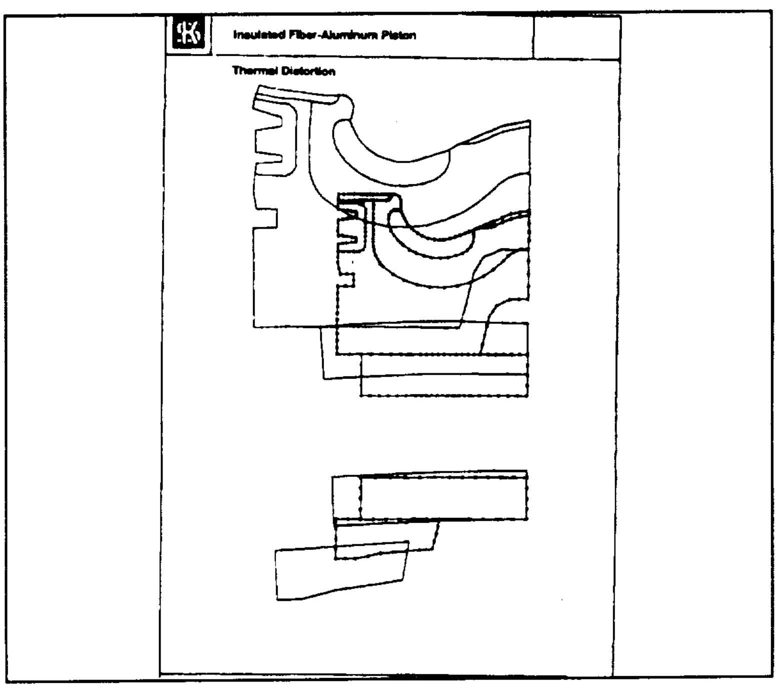


Figure 40

The rim area is especially distorted due to the interfaces between 4 different materials and high temperatures. Resulting maximum principle stresses are shown in Figure 41. The area of greatest concern is the underside of the aluminum titanate insulator with a tensile stress of 500 MPa (72.5 KSI) which is an order of magnitude greater than the strength of the material. At this stage of the analysis it was decided that the concept was not viable and no further work was done.

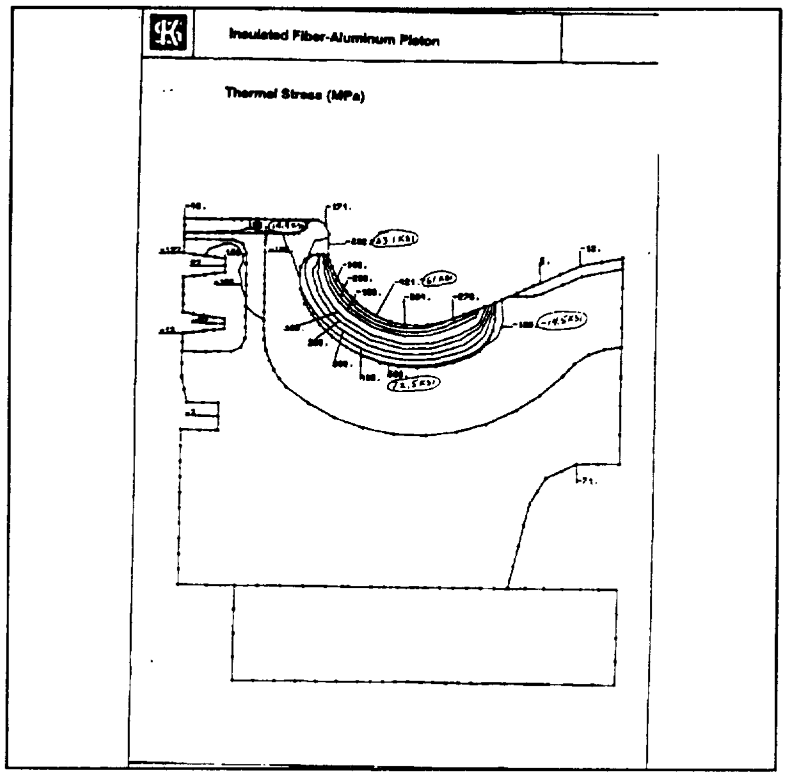


Figure 41

SPHERICAL JOINT PISTON

The third major piston design alternative investigated is unique mechanically. Instead of the typical wrist pin joint between piston and connecting rod, this concept uses a spherical, or ball and socket type joint. Spherical joint pistons are occasionally used on large bore, low speed diesels and are used on very small engines such as those used in model airplanes. Use of a spherical joint allows the piston to rotate in the cylinder bore.

This will reduce the thermally induced stresses generated by temperature gradients from injector spray plumes and intake/exhaust stratification. The entire piston is basically axisymmetric which yields uniform thermal growth at operating temperatures. This feature should allow for a better fit between the piston and bore for better guidance and improved oil consumption. One manufacturer of low and medium speed diesels reports that a recent change to a spherical joint piston in one of their models results in decreased oil consumption. Cylinder pressure capability is good for the spherical joint design because of the large bearing area possible along with the uniform support that avoids bending around the pin axis.

Cummins took the lead in generating spherical joint design concepts with evaluation and refinement of the designs a joint Cummins/KS effort. Single and multi piece designs using a variety of materials and assembly methods were investigated. Many of the early designs utilized iron or steel in all or part of the piston. While a ferrous piston is attractive for its high temperature material properties, sections need to be thin due to the low conductivity and high density of this material. This leads to expensive casting, forging, and machining processes. An additional drawback is inflexibility in bowl shape. Iron or steel pistons are not generally good choices for deep or reentrant type bowl shapes because of thermal fatigue at the bowl rim caused by low conductivity. Also because of the requirement for thin sections, once a bowl shape is chosen, it cannot be modified for development purposes. The decision was made to concentrate on a fiber reinforced, squeeze cast aluminum piston material for its light weight, good conductivity, and thermal and mechanical fatigue strength.

The design that evolved from the concept studies was then evaluated by finite element analysis. Because of the symmetrical nature of the design, a 2-dimensional model was used as shown in Figure 42. The 2-D model allowed a fine mesh to be used for accuracy without leading to long computer run times. Several material zones exist in the design as illustrated in Figure 43. The base material is squeeze cast aluminum and is in the bearing and skirt areas of the piston. The upper portion of the piston is reinforced with ceramic fibers. The chosen fibers are Saffil reinforcement grade fibers made of 95% aluminum oxide and 5% silicon oxide. Several alternatives were investigated for reinforcement of top and intermediate ring grooves including powdered metal, fiber and particle reinforcement, and cast-in niresist. In this model a niresist ring carrier was chosen as its properties are best understood. To lower heat rejection and increase the durability of the underlying aluminum matrix material, a plasma sprayed thermal barrier coating is applied to the crown. In this first iteration, a 2.5 mm thick (.100 inch), three layer coating such as has been developed on DEN3-331, Thick

Spherical Joint Piston

Finite Element Model

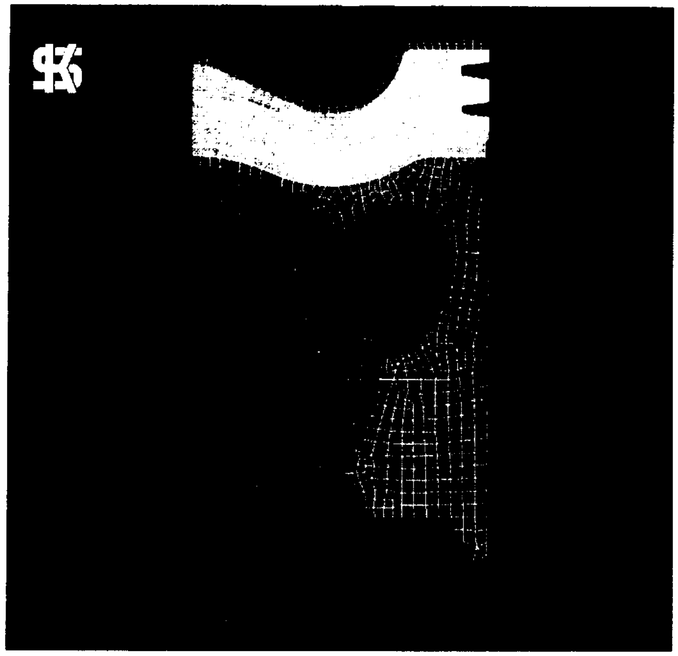


Figure 42

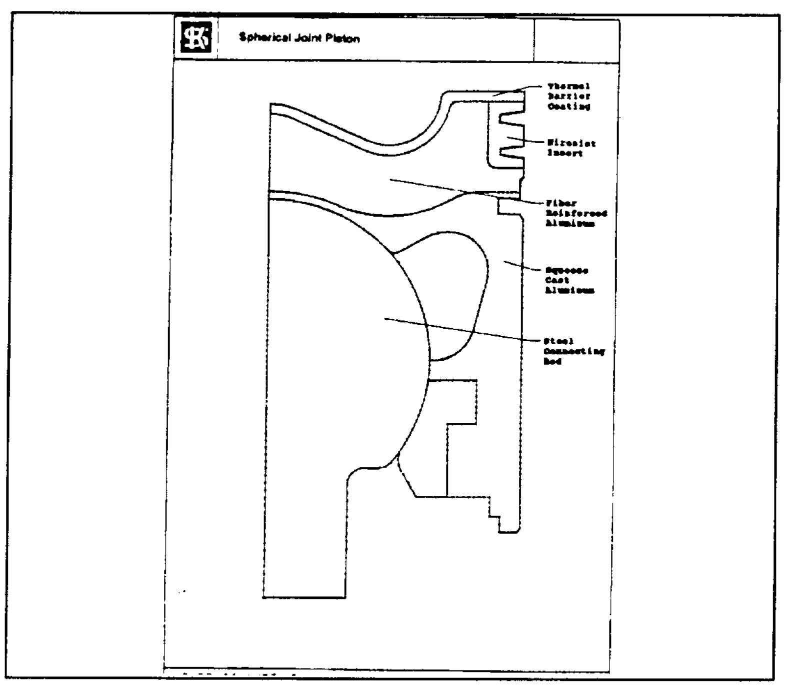


Figure 43

Thermal Barrier Coatings was used. No active spray cooling of the underside of the piston was modeled although lubricating oil fed through the connecting rod is assumed to provide a small amount of cooling.

Predicted temperatures throughout the piston are shown in Figure 44. The thermal barrier coating is effective in keeping material temperatures at reasonable levels in spite of the lack of active cooling. In the ring groove and ball/socket interface the temperature is high enough that good durability would be questionable without improvements in lubricants such as are under development in contract DEN3-373, High Temperature Liquid Lubricants. Maximum surface temperature on the thermal barrier coating is 775 C (1427 F) at the bowl rim. Maximum in the fiber reinforced area is 268 C (514 F) while the maximum in the unreinforced aluminum is 241 C (466 F). All temperatures are within a generally acceptable range for the materials chosen. The temperature distribution causes thermal distortion as shown in exaggerated form in Figure 45. Due to the symmetrical design, the distortion is uniform and predictable. The largest concern is in the lower ball/socket joint where the growth of the skirt causes the joint to open up at full load operating temperatures. At top dead center during the non-firing stroke, the inertia load would cause an impact when the clearance is taken up. The lower part of the joint is not designed to take such large loads. A secondary concern is the distortion of the top ring groove caused by the mismatch of thermal expansion coefficients. Both of these concerns can be addressed in optimization of the design. The stress induced by the thermal distortion is shown in Figure 46. Maximum principle stress is shown in MPa. The thermal barrier coating (TBC) is almost entirely in compression which is critical for the plasma sprayed ceramic to survive. The TBC/ring groove insert interface is cause for some concern as the distortion induces tensile stress in the coating. However, this is in the bottom layer of the coating containing a large percentage of metal and is within a range acceptable to the CoCrAlY material.

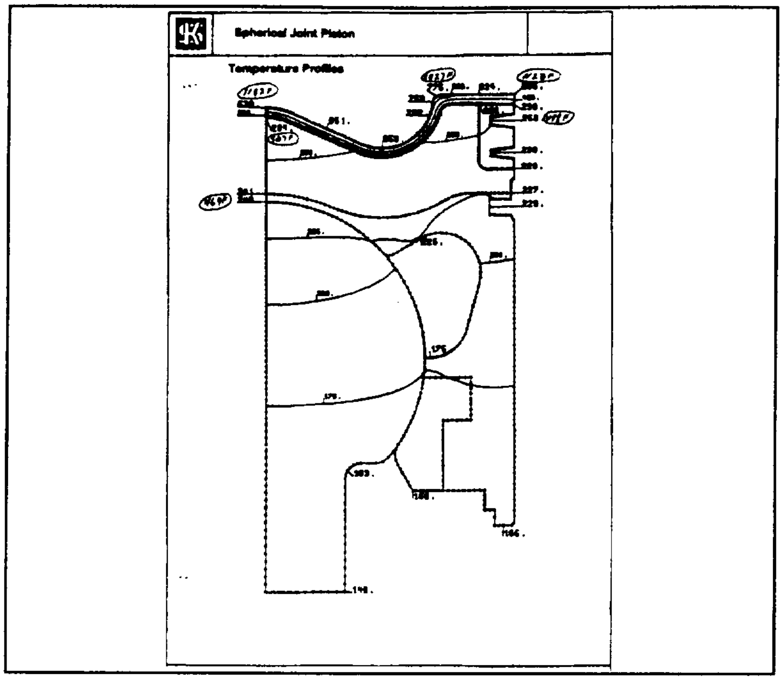


Figure 44

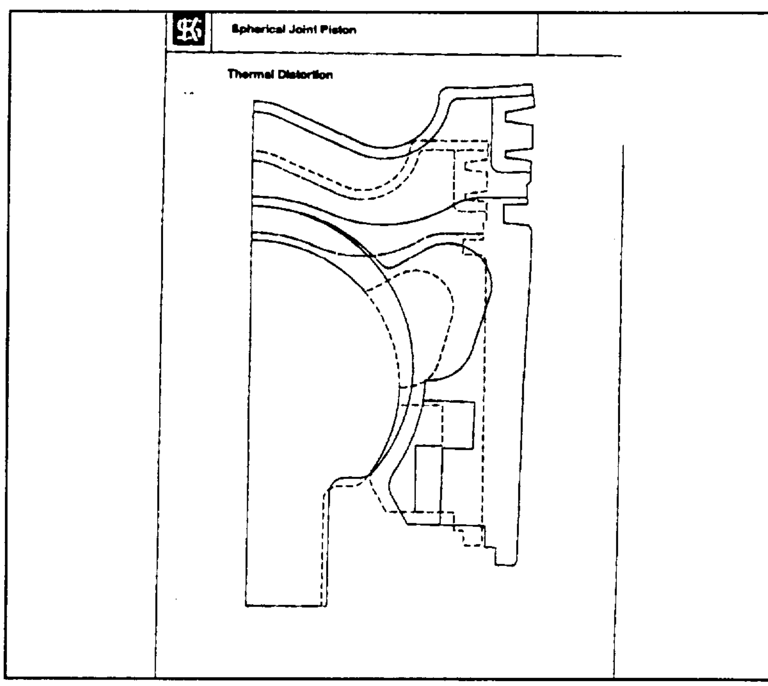


Figure 45

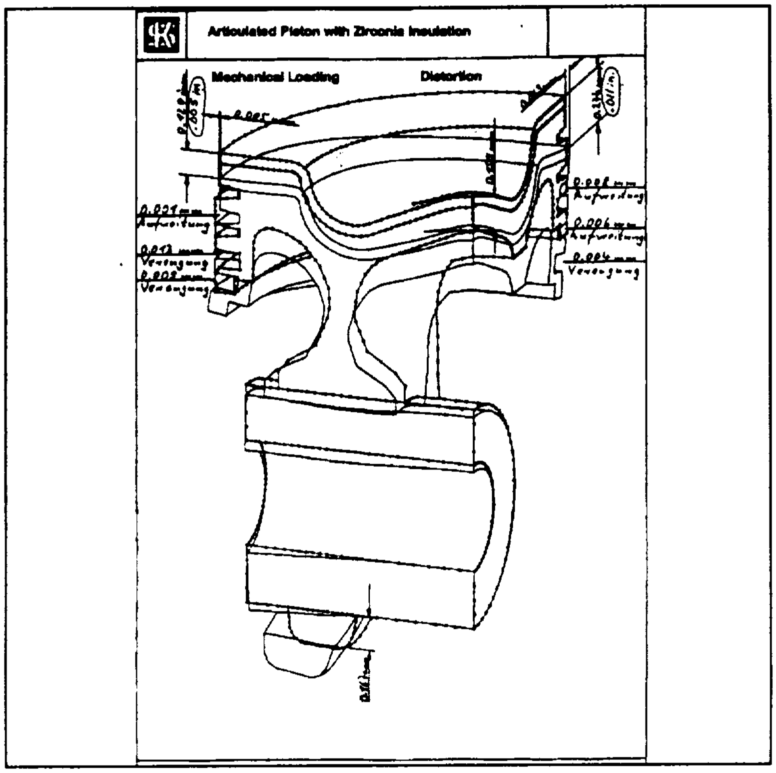


Figure 46

When the mechanical load of 3000 psi cylinder pressure is added to the thermal load, the distortion is as shown in Figure 47. The symmetry of the design and large bearing area avoid any radical deformations. The lower portion of the spherical joint opens up more under this condition and would release any lubricating oil trapped in the annular groove, however immediately upon lowering of the cylinder pressure during the expansion stroke, the gap narrows to that induced by thermal loads. Predicted maximum principle stress from the combination of thermal and mechanical loads is shown in Figure 48. Areas of concern include the edge of the ball/socket interface and the ball/shank fillet on the connecting rod. Up to this point the FE analysis has not included an oil film between the ball and socket. Inclusion of the actual film will tend to spread out the loading and avoid concentration of the load as shown in the plot. Also, both the ball and socket are modeled as perfect spheres. The plan for the final design is to use a spherical ball and adjust the contour of the socket surface to create uniform loading in the joint. The connecting rod detailed design and analysis will be Cummins' responsibility and will be addressed in Phase 2.

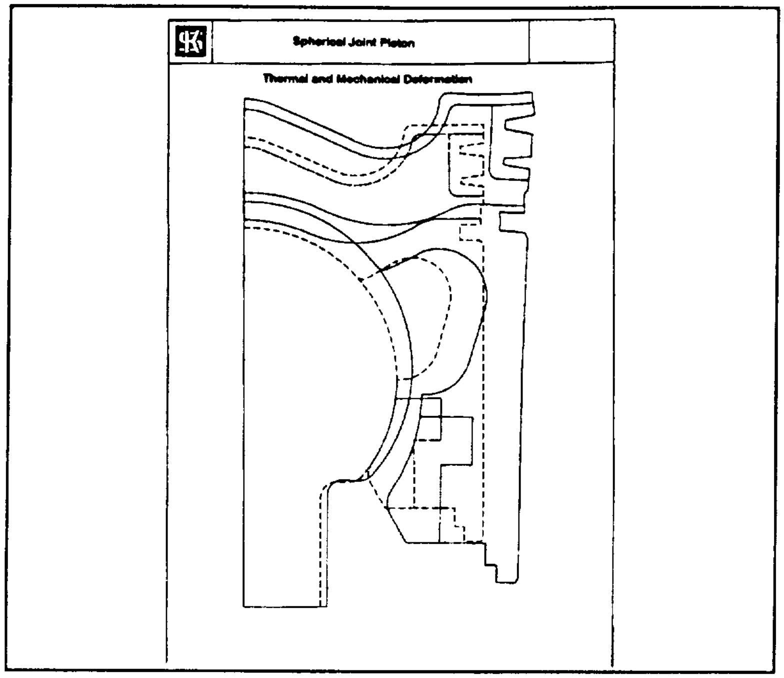


Figure 47

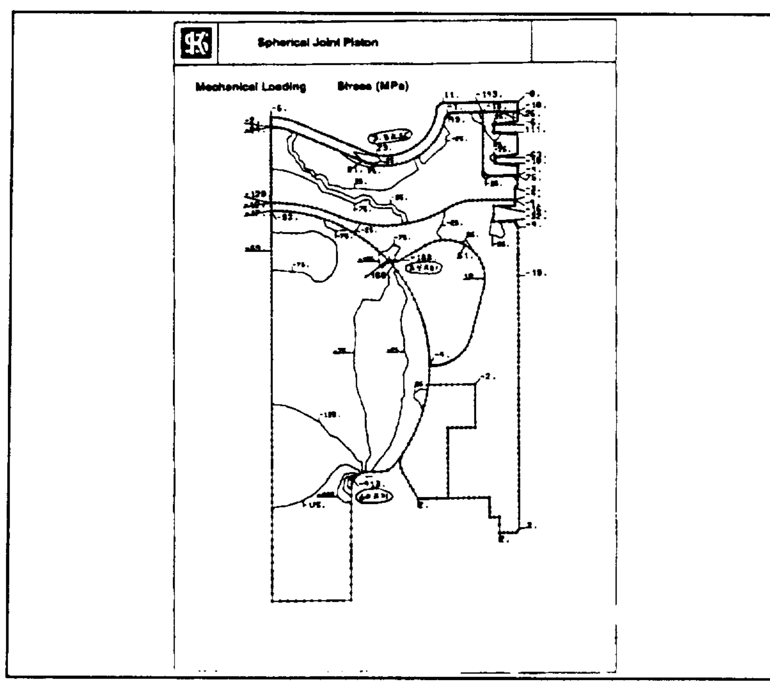


Figure 48

HEAT TRANSFER ANALYSIS

An analysis of heat transfer with the various piston design concepts was performed using the FE models. Material properties were adjusted so that the effects of the ceramic insulation could be evaluated. Figure 49 shows the distribution of heat transfer within the articulated piston design, with and without the zirconia thermal barrier coating. Using 100% as the total heat transfer through the uninsulated piston, the heat through the piston ring grooves accounts for 30%, 15%, and 8% of the total. The largest amount, 43%, is transferred through the inner surface to the crankcase oil mist. This number would be much larger if active oil cooling of this surface were included. Only 4% goes through the piston pin and it is assumed that there is no direct thermal connection between the crown and skirt of this piston design. When the TBC is added to the top of the crown, the total heat transfer drops to 54% of the uninsulated value. The distribution is similar to the uninsulated case.

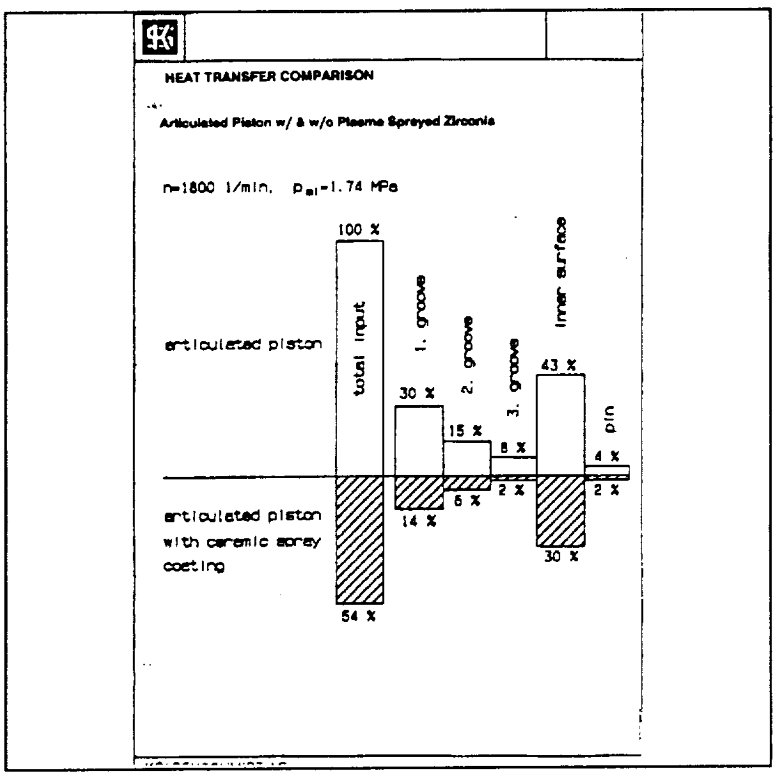


Figure 49

The aluminum piston was analyzed with and without the cast-in and sprayed-on ceramics in the design. The non-insulated version is a conventional aluminum design and its heat transfer is shown in Figure 50 as 100%. The distribution is different from the articulated piston due to fundamental design differences creating different heat flow paths. The heat through the ring grooves is 20%, 15%, and 11%. The inner surface accounts for 20% of the total, the pin 20%, and the skirt 20%. Compared to the articulated design, the one-piece aluminum design has a much better thermal connection to the skirt and pin but less inner surface area. The addition of the ceramic insulators decreases the overall heat transfer to 62% of the baseline with the distribution in similar proportions to the uninsulated case.

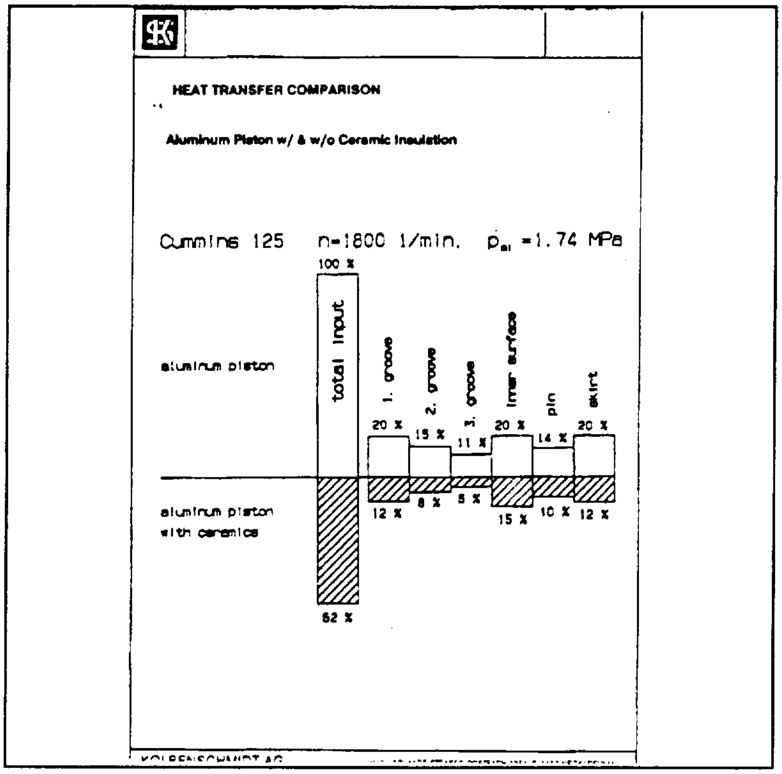


Figure 50

All four combinations are compared in Figure 51. Here, the heat transfer through the uninsulated aluminum piston is taken to be 100%. The uninsulated articulated piston shown at the bottom, has a heat transfer of 81% of the baseline. This shows the combination of the lower conductivity steel crown plus the thermally isolating design of the articulated piston result in lower heat transfer compared to an aluminum design. The typical practice on current engines to increase the oil cooling on articulated pistons to increase heat transfer and reduce metal temperatures illustrates this fundamental difference. One observation from this thermal analysis is that heat transfer and resulting metal temperatures in different parts of the piston can be manipulated by both material and design changes. Both tools will be used in design optimization in Phase 2 of the program. Time did not allow a heat transfer analysis of the spherical joint piston.

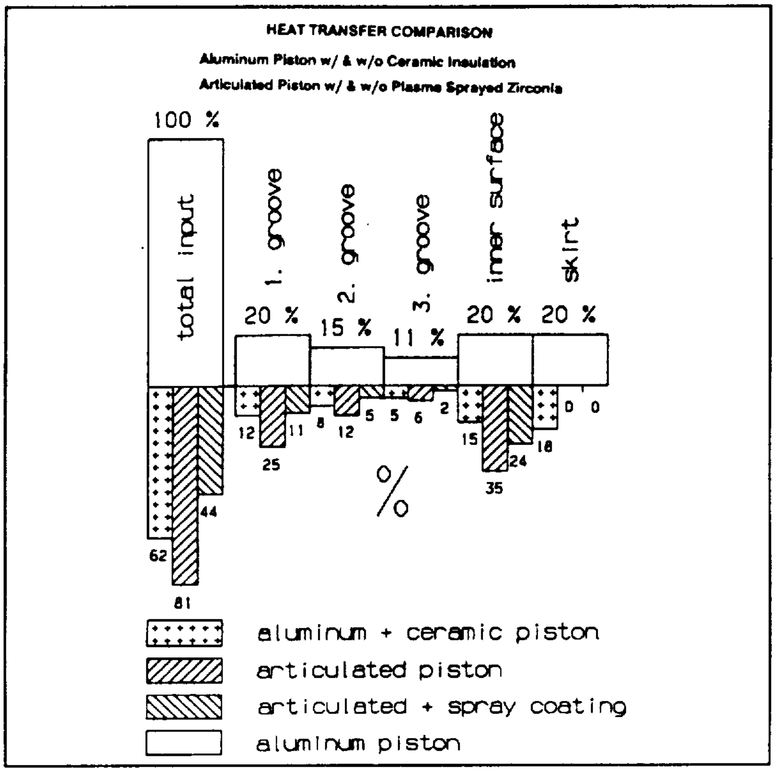


Figure 51

PISTON CONCEPT SELECTION

After weighing the relative merits of the three basic piston design concepts, it was clear that the spherical joint design provided the most opportunities to solve many of the fundamental piston design problems facing future engines. While an iron or steel version may have good chances of success if a conservative bowl shape is chosen, the bowl shape flexibility, weight, and cost of fiber reinforced aluminum make it attractive. The final selection of a fiber reinforced aluminum spherical joint piston was made with the understanding that there is technical risk involved and much development required. The potential payoffs are seen as being great enough to warrant choosing this approach.

As a backup approach, investigation will continue into an articulated piston made from nickel aluminide. A low heat rejection piston could be made of this material without use of thermal barrier coatings, although surface temperatures would strain the capabilities of the lubricant. As casting and forging alloys and processes are developed through association with Oak Ridge National Laboratories, a program will be laid out for the development of the Ni_3Al piston concept.

V. COMPONENT DESIGNS

Development of advanced components is the central focus of this program. The major deliverable of Phase 1 is the determination of the appropriate component concepts to be further developed and demonstrated in the next phases. The scope of "in-cylinder components" was determined to include piston, rings, connecting rod, cylinder liner, and cylinder head for this program. These are the components that make up the combustion chamber and are the thermal and thermodynamic heart of the engine. The specific components being designed are based on the production Cummins L10 engine. This will enable us to begin with existing modern designs in many cases and will mean that test engines will be readily available at the appropriate time. Figure 52 is a cross section of an L10 engine with the new In-Cylinder Components features inserted. The highlights of the design concept include an oil cooled cylinder head with insulated intake and exhaust ports. Oil cooling allows elimination of the water pump, filter, and oil to water cooler. Removal of water jackets in the cylinder heads reduces cost and opens up room for the port insulation. Insulated exhaust ports lower heat rejection, reduce metal temperature, and increase usable energy to the turbocharger. Insulated intake ports avoid reheating of the charge air before it enters the cylinder. This lowers fuel consumption and generation of nitrous oxide emissions. The cylinder liner is also oil cooled for system simplification reasons. The combustion seal between the liner and head is accomplished with a "radial combustion seal". This new seal design creates the high sealing loads necessary for high peak cylinder pressure while reducing the required head bolt load. Cylinder bore distortion is improved which will lead to lower oil consumption and therefore lower particulate emissions. A high conformability ring pack is included so that the rings are able to better follow the bore walls resulting in improved control of blowby and oil consumption. The piston uses a spherical joint with the connecting rod and advanced materials to accomplish high thermal and mechanical load capability. Some of the components will now be described in more detail.

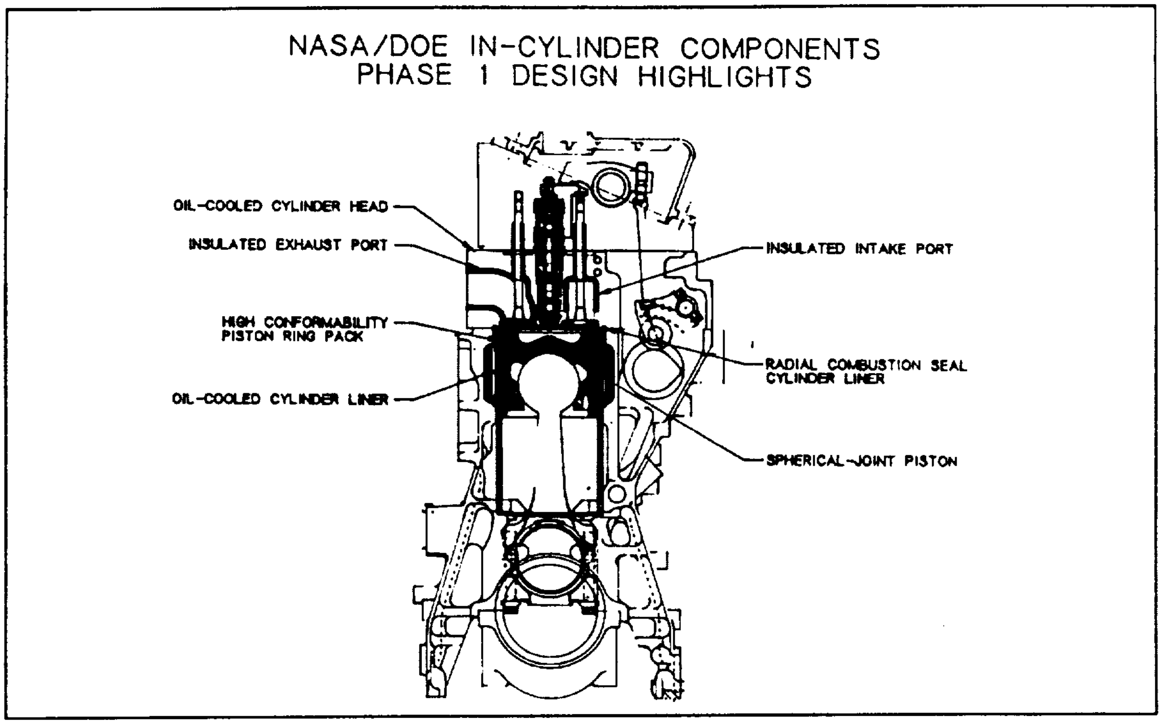


Figure 52

A. SPHERICAL JOINT PISTON

The most outstanding feature of this piston is the spherical joint concept. Use of a spherical, or ball and socket joint allows for an axisymmetric piston that is free to rotate on the connecting rod. This results in uniform temperature distribution and thermal growth. Cylinder pressure capability is high due to the large bearing area and symmetrical support. A compact piston with high mechanical and thermal load capability is possible. A sketch of the piston design as it stands at the end of Phase 1 is shown in Figure 53. The base material is squeeze cast aluminum with ceramic fiber reinforcement in critical areas. This material gives a durable, light weight piston at reasonable cost. Squeeze casting opens up many options for ring groove reinforcement. Sintered steel is an option that has been investigated at some piston manufacturers that gives excellent bonding to the base material and high strength and wear resistance. Insulating the crown of the piston with a plasma sprayed zirconia thermal barrier coating lowers temperatures in the base material without suffering excessive heat transfer to the coolant. Thermal-mechanical fatigue is therefore greatly reduced for improved piston durability. A comparison of the spherical joint piston to the conventional L10 piston is shown in Figure 54. The design shown has a shorter overall height allowing a longer connecting rod while maintaining high load capability.

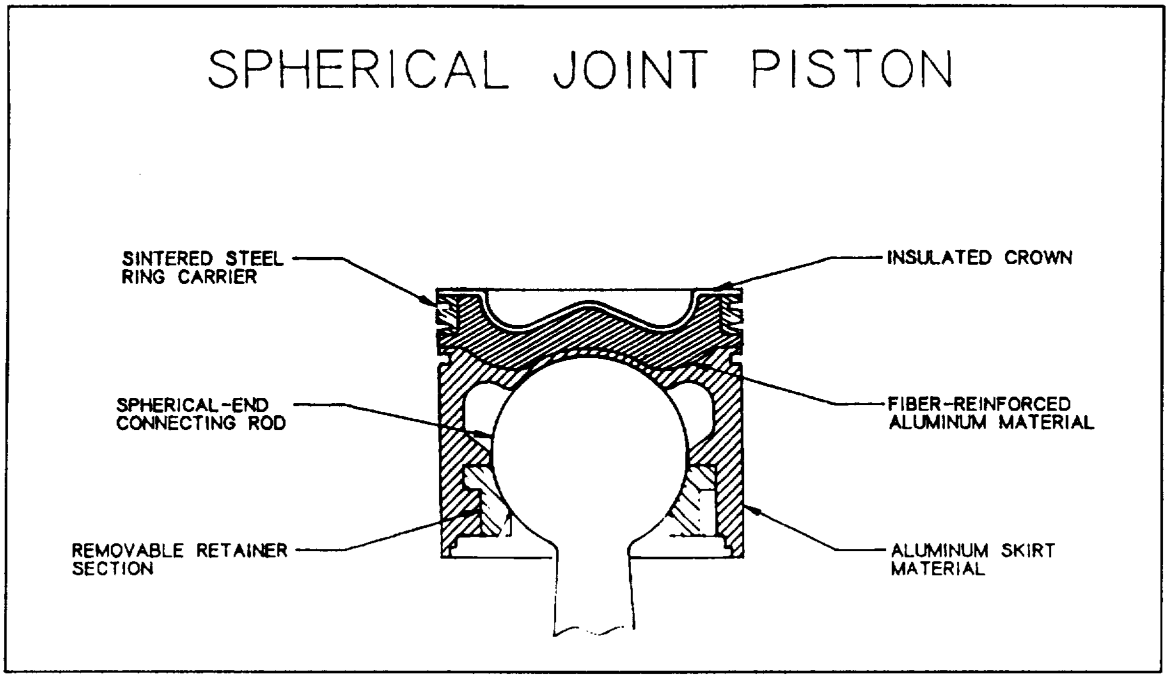


Figure 53

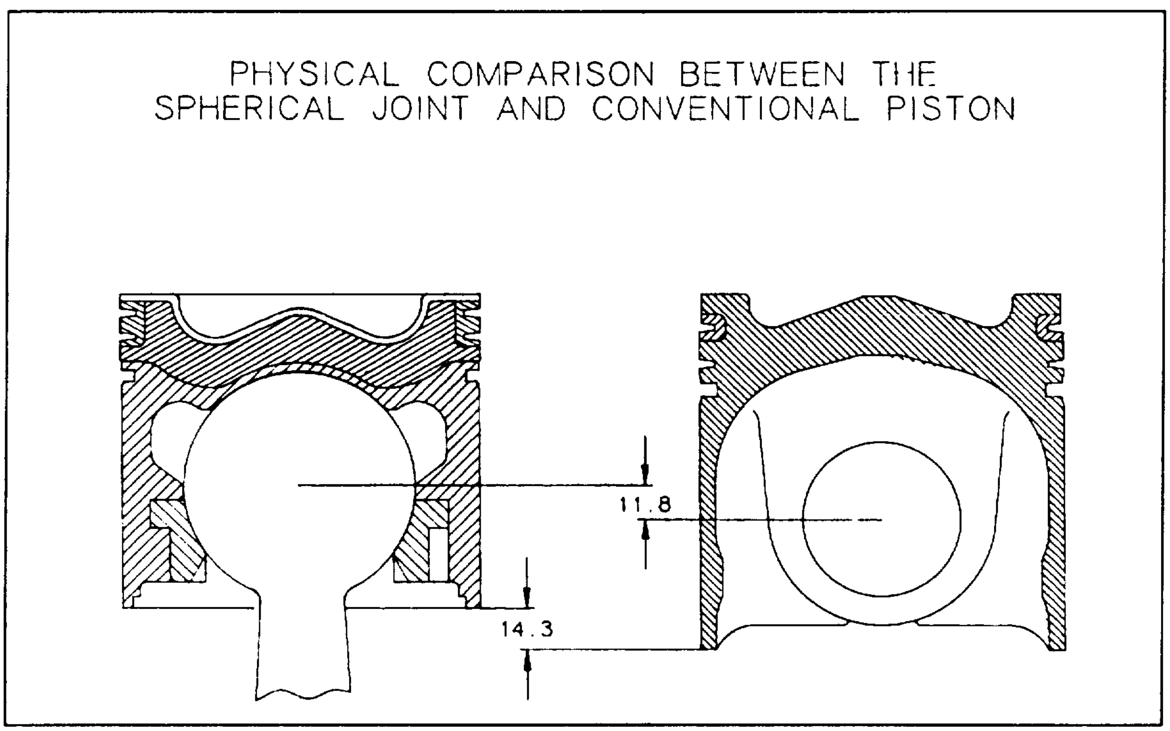


Figure 54

B. CYLINDER LINER

Compelling reasons exist for cooling the cylinder liner of a diesel engine. A cool liner improves engine volumetric efficiency for better air flow. Low temperatures in the ring/liner interface reduce wear and increase durability of piston rings and cylinder liners. Since the period of highest heat flux occurs when the piston is close to top dead center covering most of the liner, insulation of the liner walls has little direct effect on combustion chamber heat transfer. Oil film viscosity is affected by liner wall temperature and directly influences piston/ring/liner friction and wear. Moderation in wall temperature is a key to balancing low friction and low wear. To achieve suitable liner temperatures without using water as a coolant, oil cooled liners with heat transfer enhancement were chosen. A turbulator such as shown in Figure 55 will be placed in the cooling jacket between the cylinder block and liner. Ideally, the turbulator will increase the heat transfer coefficient by breaking up the laminar boundary layer and also increase the heat transfer area by creating fins. Development of a suitable turbulator design is planned in Phase 2.

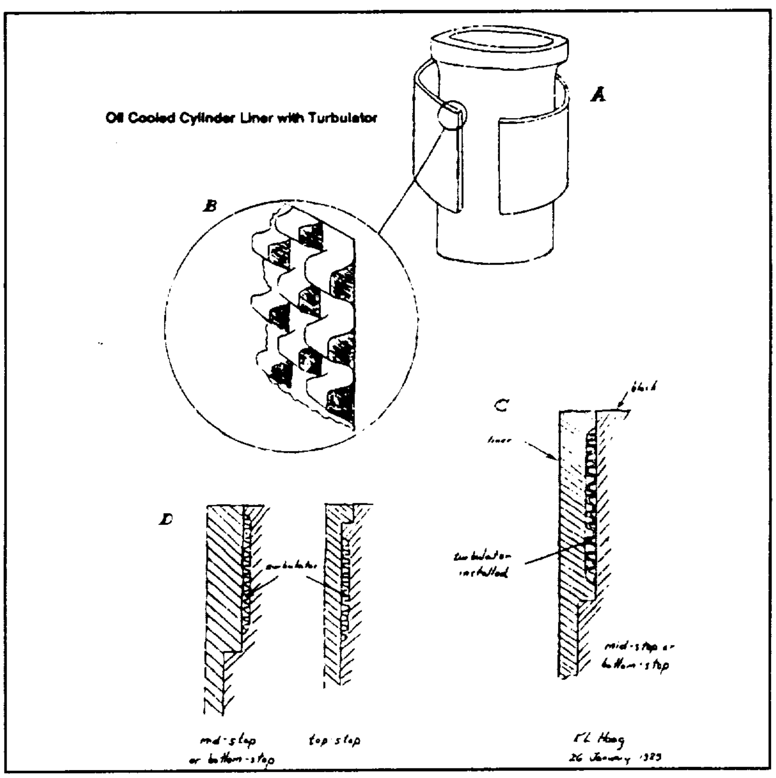


Figure 55

Distortion of the cylinder liner is a detriment to oil consumption control. Much of the distortion is caused by the clamping loads typically applied through the head gasket to effect a combustion seal. This problem is compounded when the higher cylinder pressures assumed for advanced engines need to be sealed. Making the cylinder liners an integral part of the cylinder head is one way of maintaining a combustion seal at high cylinder pressure. At least one currently produced locomotive engine has the liner and head welded together. Several concepts were conceived and evaluated for this program to provide a durable liner to head combustion seal while not generating undue distortion. One concept requires the liners to be threaded into the head. It was decided that machining, assembly, and service drawbacks made this an unattractive option.

A unique sealing concept was designed which has the ability to seal high cylinder pressure without distorting the liner. The required cylinder head bolt clamping load is lowered with this radial combustion seal concept. Instead of the sealing load being applied axially down the liner, the load is applied radially outward as shown in Figure 56. An annular groove is machined into the cylinder head face. The top surface of the liner extends into this groove, making line-to-line contact with the outside of the groove. On the liner inside diameter, a wire sealing ring is captured to perform the sealing function. This ring can be made of annealed steel wire as is commonly found in head gasket combustion seals. A slight angle on the inside surface of the groove applies load to the sealing ring with a wedge action as the cylinder head is installed. Combustion pressure on the bottom of the ring will force it further up into the angled groove, increasing the sealing force and making this a self energizing seal. Cylinder head bolt load to contain this joint is roughly half of a traditional head joint. Since no forces or moments are applied to the cylinder liner below the head surface, distortion can be minimized. Since the combustion sealing function of the head gasket is now performed with another part, the fluid sealing functions can be replaced with o-ring type seals, eliminating the head gasket altogether. The resulting metal to metal joint between the block and head is more structurally sound than a gasketed joint.

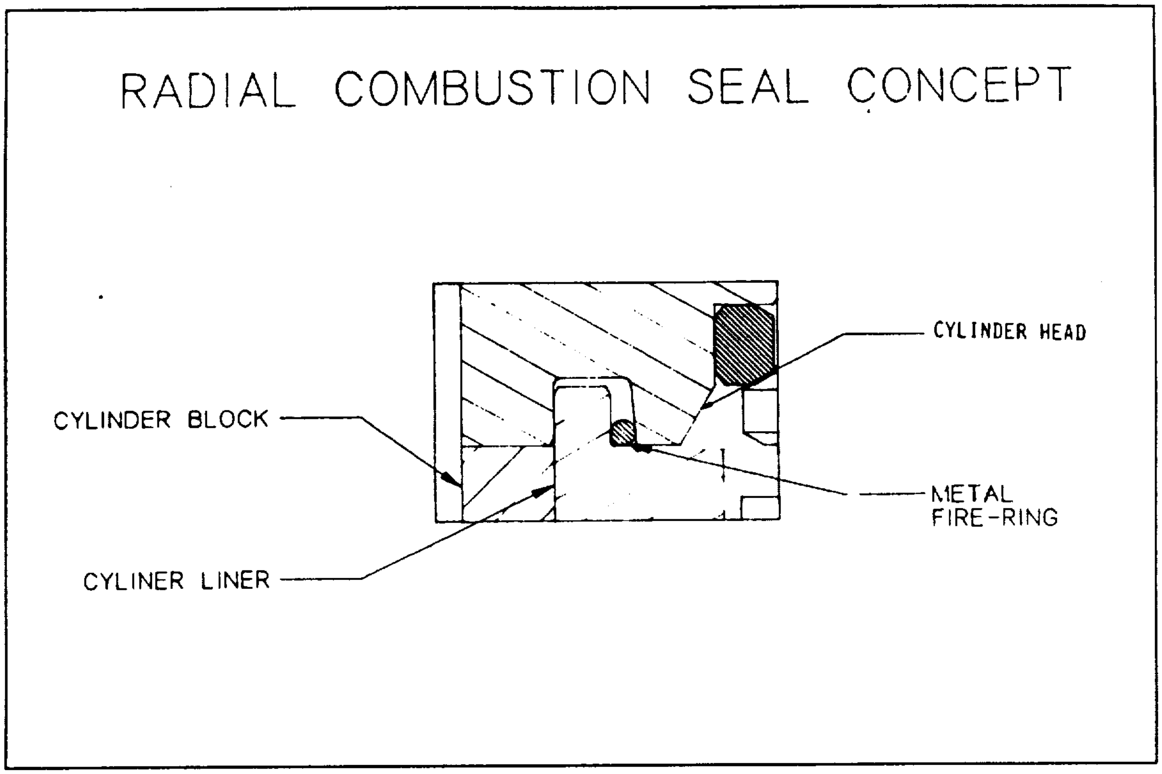


Figure 56

C. PISTON RINGS

Piston rings are critical to the advanced power cylinder. The primary functions of piston rings are to seal combustion from the crankcase and to seal lubricating oil from the combustion chamber. As stated previously, control of oil consumption is critical to meeting future particulate emission regulations. Higher cylinder pressure will require improved top ring performance in both sealing and structural durability. High BMEP will raise temperatures and thermal loads. Current diesel piston ring technology is primarily centered around use of ductile iron rings with electroplated chromium coatings. Even at higher output levels, this technology should remain sufficient for intermediate and oil rings. The top ring is therefore the area to concentrate new material and design efforts. Compliance to cylinder wall irregularities has been shown to improve ring performance, and as such is a design criterion for this program. The likely choice of ring base material is a steel with high chromium and molybdenum content. The surface coating will likely contain ceramics for wear resistance.

Combustion Technologies, Inc., a subsidiary of Cummins, is responsible for the details of ring design, analysis, and prototype manufacturing. They also possess plasma spray capabilities for advanced coatings. Ring coating direction will closely follow the lead of the Wear Resistant Ceramic Coatings program (ORNL contract 86X-SA581V) currently underway.

D. CYLINDER HEAD

As was discussed in the thermal analysis section, the proposed cylinder head is of grey iron material with strategic oil cooling. There will be no cast-in cooling jackets. Cooling will be performed locally with oil drillings. In order to enhance the heat transfer, turbulators will be developed that will insert into the drillings. To reduce heat rejection and base material metal temperatures, ceramic insulating exhaust port liners are being incorporated. A small, but measurable performance improvement is available from the added exhaust energy to the turbocharger and turbocompound system. The intake ports are also insulated with ceramics to avoid heating the intake air before it reaches the cylinder. Reheating the intake air after it has been drastically cooled by an advanced aftercooling system would be counter productive to good fuel consumption and low NOx emissions. The need to insulate the intake ports is increased in the strategically oil cooled cylinder head due to higher average metal temperatures. Cummins has demonstrated successful casting of ceramic ports into grey iron on two different Defense Department sponsored research programs and it is felt to be a viable, if risky option for future engines. Figure 57 is a section view through the cylinder head concept drawing showing the location of the oil drillings and the insulated ports. The valve and capscrew locations are unchanged from the production L10 cylinder head. Based on analysis of a similar head, it was not necessary to cool between the two intake valves. Figures 58 and 59 show more of the port design concepts. By eliminating the coolant jackets, there is now room for the ceramic port liners as well as for larger exhaust exit area for improved aerodynamics. The valve bridge area on the combustion face, especially between the two exhaust valves is the area of greatest concern for thermal-mechanical fatigue. Past experiments at coating this area with plasma sprayed ceramic insulation have often failed due to the geometry. Ceramic insulating head plates have not proven to be robust or cost effective. Low conductivity metal head plate concepts raise concerns about cost and sealing. A new approach has been chosen that addresses the fundamental valve bridge fatigue problem, while preserving the general grey iron structure. A metal valve bridge insert on the combustion face thermally isolates the grey iron from the high combustion temperatures. The insert is shaped roughly like a cross, with each leg being a valve bride. This insert could be cast in place or installed in a machined pocket. The primary design at this

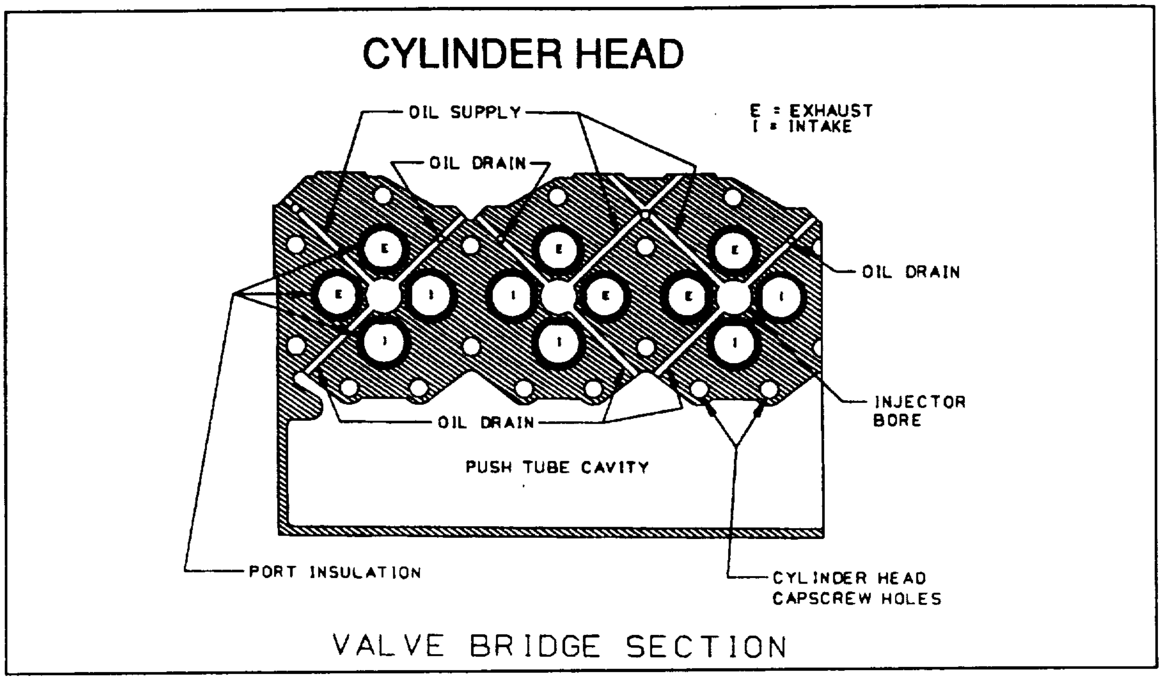


Figure 57

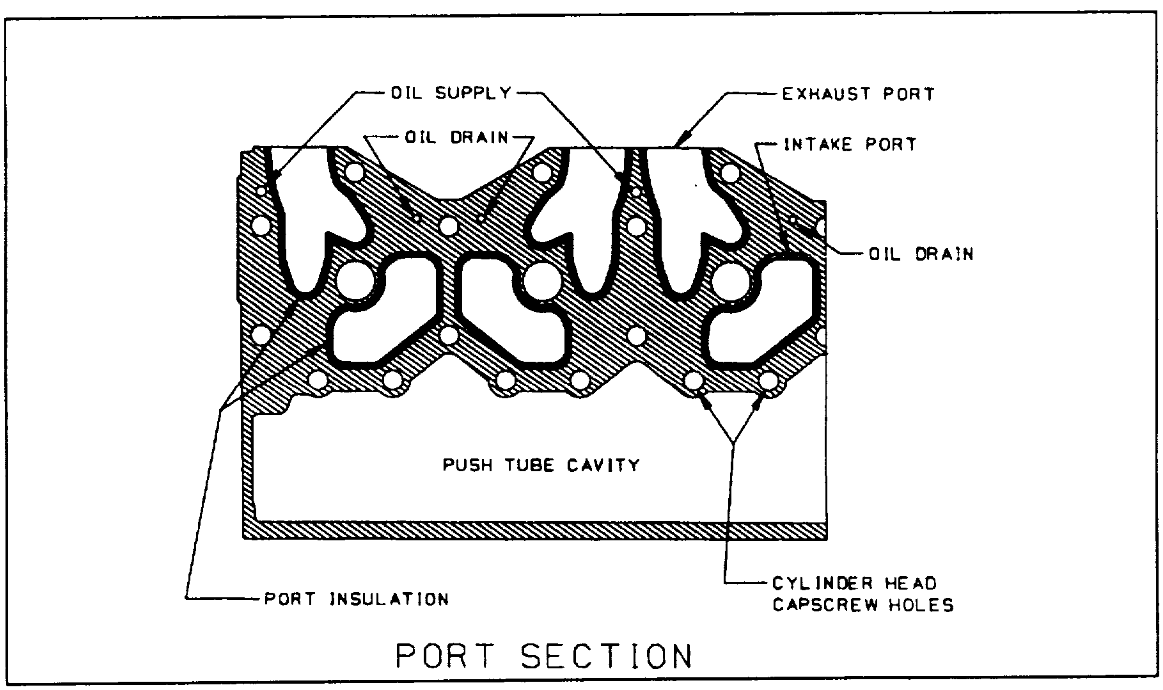


Figure 58

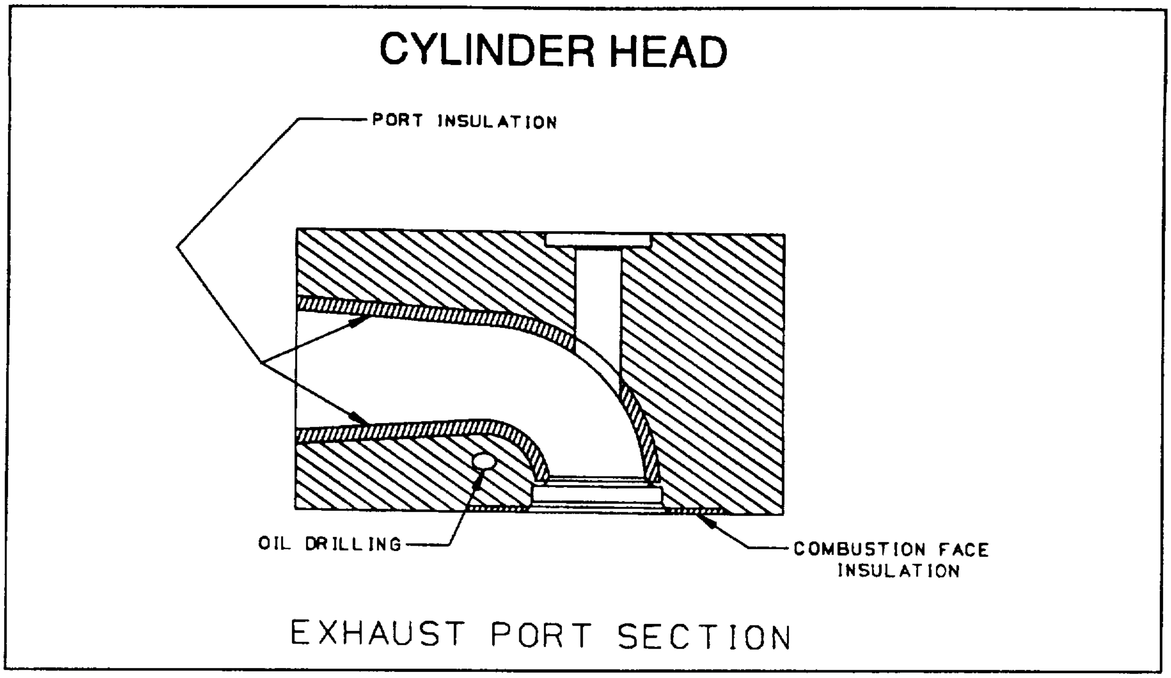


Figure 59

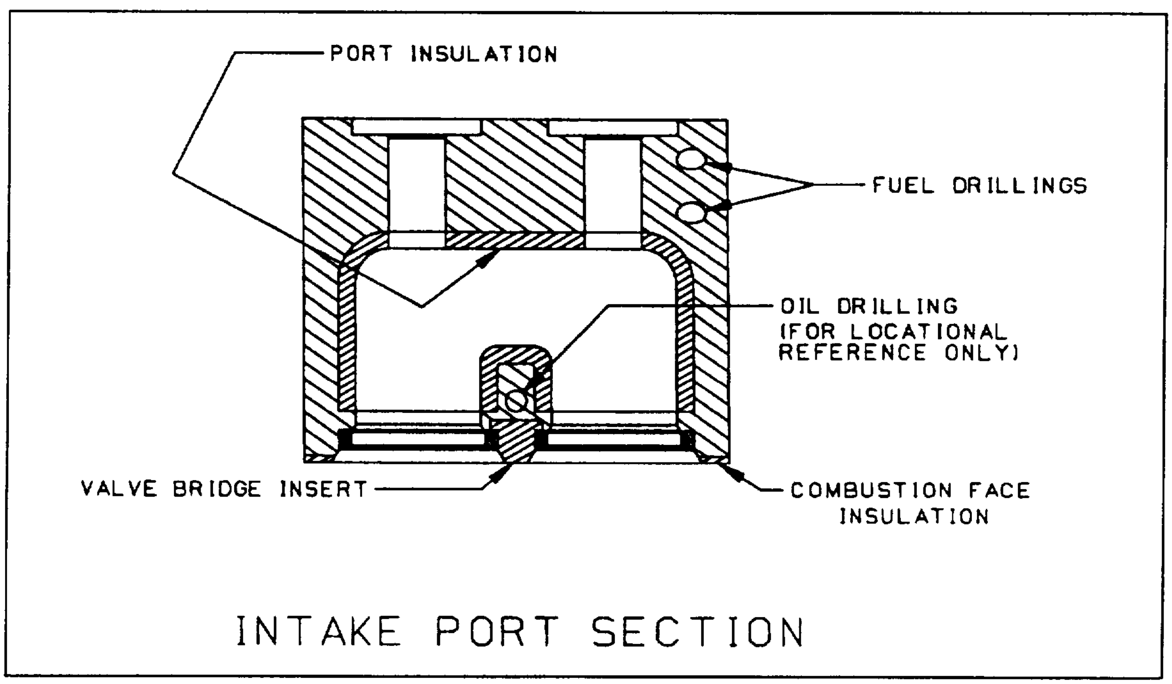


Figure 60

time is to install the fully machined insert such that there is clearance for it to expand radially when heated to avoid building up compressive stress. The combustion face valve bridge insert is held in place by the four valve seat inserts. Material possibilities for the insert are numerous, with stainless steel or nickel aluminide being preferred at this time. The outer perimeter of the combustion face is insulated with plasma sprayed zirconia for additional heat rejection reduction. Figure 60, a cross-section through the center of the intake valves, shows the position of the valve bridge insert and the zirconia insulation.

In Phase 2, a detailed finite element analysis of the cylinder head will be performed in order to evaluate the effectiveness of the design in controlling heat rejection and thermal fatigue. Each feature will be evaluated independently to assess its individual merit.

VI. ECONOMIC ANALYSIS

The challenge for the developer of commercial diesel engines is to find ways to implement advanced technology and to offset any necessary cost increase with a cost reduction to the customer so that in the end the customer does not have to pay extra for new technology. The most direct way to recover a cost increase is through an associated cost reduction by design. An example is insulated ports in the cylinder head. Ceramic ports are an additional cost. However, if all water jacket and port cores are removed, the overall cost does not change drastically. Removal of the water pump, filter, and oil to water cooler will also reduce costs directly to cover increases in other areas. Sometimes the cost reduction is less direct. A reduction in fuel consumption is a direct cost savings to the customer. Fuel costs are the largest operating cost to the majority of heavy duty truck operators and are weighed against first cost of the engine. In order to be competitive in the marketplace, there should be payback on the initial investment in a year or less. Increased reliability and durability are harder to quantify, but are just as important as fuel consumption. The greater the amount of time an engine is in revenue producing operation, the greater the value to the owner. Reductions in emissions are necessary to have a viable product and as such are a cost of being in the engine business. There is value to society in lower emissions that all will benefit from. The engine manufacturer must choose the most cost effective way to meet the emissions requirements to best serve society and the direct customers.

For this program, a simplistic, but accurate cost goal was used. Any cost increase must be accompanied by a proportional increase in power output potential so the dollars per horsepower ratio does not change. Heavy duty diesel engines are not generally sold by displacement, but by the amount of work they can do (horsepower). Assuming that durability can be maintained, a customer will be just as satisfied with a 10 litre 400 horsepower engine as a 14 litre engine. Uprating an engine will often require additional costs in airhandling and fuel systems, so internal component costs must increase proportionally less to become cost effective.

In Table 8, the major component design features are listed with + and - to signify whether the cost is expected to increase or decrease compared to a state of the art diesel engine such as the 1989 production L10. In the benefit column, the advantages of each feature that result in increased value to the customer are listed. As a total package, it is felt that the cost/benefit ratio of the concepts in this program is less than 1 which justifies the additional costs associated with the new technologies. Based on preliminary analysis, the concepts shown in this report meet the goal of constant $/hp.

VII. CYLINDER KIT DYNAMICS MODEL

The piston/ring/liner interface is very dynamic during the 4-stroke cycle. The rings are contained by a moving piston, follow a non-perfect cylinder, and are subjected to varying pressure and inertia loads. An analytical study was performed to identify areas of opportunity for improvement of oil consumption and blowby on the L10 engine and also to evaluate the usefulness of a cylinder kit model available from Compu-tec Engineering. A consultant, Harold McCormick was retained to construct the input files, interact with Compu-tec, and interpret the output. The major focus was on particulate emission reduction through lower oil consumption. The components chosen for the study were the current production L10 or 88L10 engine cylinder kit. This piston uses three piston rings. Seven critical features were varied in a partial factorial test manner. These variables included: top and second ring gap area, second and third piston land volume, top and second ring bottom seating angles, and second ring mass. High and low values for the variables were chosen from print tolerance extremes or other suitable moderate deviations. The cylinder kit performance was analyzed at four engine operating conditions of significance to the EPA transient emissions cycle. The chosen points were 1) rated power, 2) torque peak, 3) low idle, and 4) 80% speed/60% load. Some of the capabilities and features of the model include:

- distorted liner shape from measurements
- structural model of ring/liner interaction
- ring motion and forces
- inter-ring pressures
- quantitative blowby prediction
- qualitative oil consumption prediction

Table 8

ECONOMIC ANALYSIS
PHASE 1 SUMMARY

	COST	BENEFIT
Cylinder Head	+ Port Insulation - Water Jacket Removal	+ Improved BSFC + Increased BMEP & Durability + Cooling System Simplification
Piston	+ Fiber Reinforcement + TBC +/- Spherical Joint	+ Increased BMEP & Durability + Reduced Oil Consumption
Piston Rings	+/- Steel Substrate +/- Wear Coatings	+ Higher BMEP & Durability + Lower Oil Consumption
Cylinder Liner	+ Turbulator +/- Radial Combustion Seal	+ Higher BMEP + Lower Oil Consumption + Cooling System Simplification
Oil Cooling	- Remove Water Pump - No Oil/Water Cooler	+ Lower Parasitic HP + Non-Pressurized

In order to evaluate all seven variables or factors, eight runs at each speed and load point were required, for a total of thirty-two program runs. The entire output is quite extensive, so only a few selected items will be included here. Figure 61 shows the cylinder pressure (an input parameter) and calculated inter-ring pressure at rated power for one set of variables. One interesting phenomenon is that the pressure under the top ring is higher than the pressure on top of the ring during the later portion of the power stroke and most of the exhaust stroke. This is due to the high pressure gas that leaks past the top ring during the peak cylinder pressure period must then leak out of the inter-ring volume. Cylinder pressure reduces quickly due to the expanding cylinder volume and the opening of the exhaust valve. Figure 62 shows the predicted force between the rings and ring grooves. A positive force signifies the ring is seated on the bottom of the groove, a negative force means seated on the top. A zero force signifies an unstable ring without enough force to seat it on either side of the groove. For the case shown, the top ring is generally well seated on the bottom of the groove except during the period of higher pressure below than on top of the ring during the exhaust stroke. The second ring shows more of the effects of inertia forces since the pressure load is smaller. The oil ring reacts almost entirely to inertia loads, as only very small pressures are exerted on it.

A summary of the effects the factors had on blowby is shown in Table 9. For each factor, the high and low value is shown. A positive number signifies the blowby increase in CFM that is predicted when the factor is changed from the low to the high value. A single asterisk (*) signifies statistical significance at a 70% confidence level. Two asterisks (**) signify significance at a 90% confidence level. As might be expected, increasing the top or second ring gap area resulted in increased blowby. What was not expected was that increasing the piston third land (2nd and 3rd interring) volume also increased blowby.

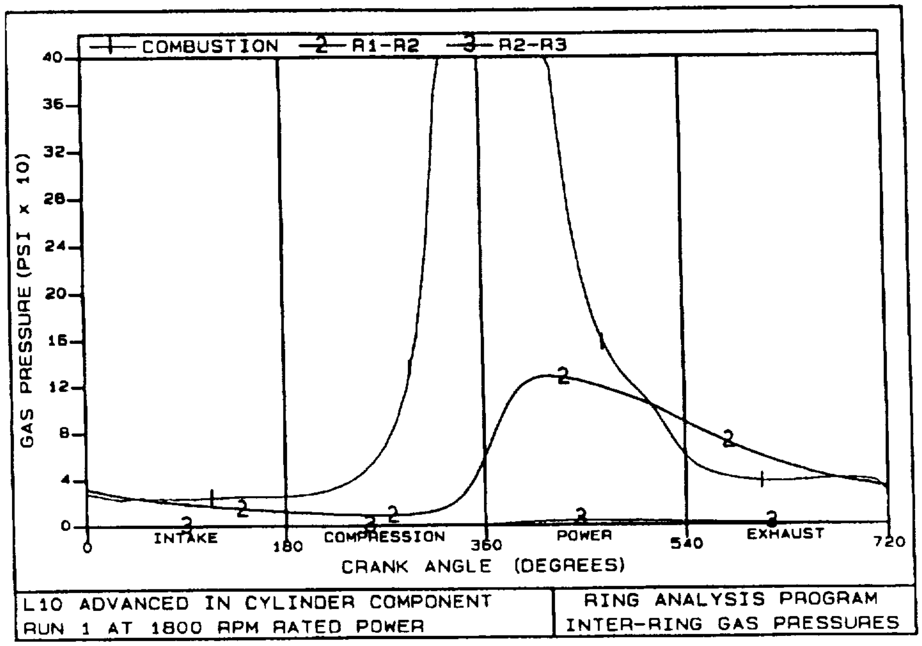


Figure 61

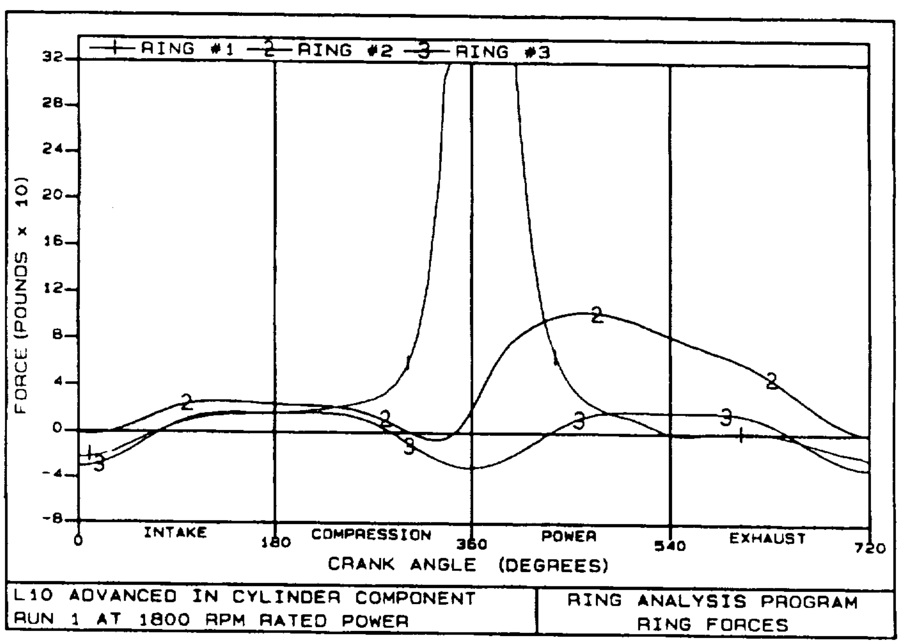


Figure 62

Table 9

Statistical Effect on Blow-by Control

Effect of Changing From Low Level to High Level
90% Confidence Level **
70% Confidence Level *

Factor Description	Low-Value High-Value	Blowby Change in CFM 700 RPM Low Idle	1800 RPM No Load	1800 RPM Rated Pwr	1600 RPM 213 HP
Top Ring Gap Orifice Area	1.8E-4 in2 5.8E-4 in2	+60**	+68**	+168**	+121**
Second Ring Orifice Area	4.5E-4 in2 14.0E-4 in2	+40**	+39**	+143**	+71**
Interring Vol Top/2nd rings	0.079 in3 0.158 in3	-5	-3	-10	-9
Interring Vol 2nd/3rd rings	0.57 in3 .114 in3	+23*	+38**	+72**	+51**
Second Ring Mass	.066 lb. .082 lb.	-4	-2	-8	-7
Side Seal Position Top Ring	Slight Front Edge Heavy Front Edge	+5	+1	+25	-2
Side Seal Position 2nd Ring	Flat Seal Rear Edge	-3	+1	-3	-2
Mean Blowby	(CFM)	71	87	205	154

Table 10 summarizes oil consumption predictions. Oil consumption was subjectively ranked from zero to ten for each run, with higher numbers meaning improved oil consumption. A positive number in the table then signifies an improvement in oil consumption when the factor is changed from the low to high value. Five results of interest may be noted. First, if the gap area is increased on both the top and second rings, the idle oil consumption will improve. This would have a positive impact on transient cycle particulates since much time is spent at idle. However, as was shown in Table 9, this would also greatly increase blowby and the two effects would have to be traded off accordingly. Increasing the 2nd land volume has a positive effect on oil consumption at both the rated and intermediate power points. No blowby increase would be expected, and both transient cycle and over the road oil consumption would be expected to improve. Increased second ring mass is predicted to improve oil consumption at 1800 rpm no-load and the intermediate point. At these two operating points the inertia load can dominate the pressure load which makes the ring mass important. The last conclusion of interest is that more front edge sealing on the lower side of the top ring is helpful at 1800 rpm/no load, but makes things worse at the intermediate point.

Table 10

Statistical Effect on Oil Control

Effect of Changing From Low Level to High Level
90% Confidence Level **
70% Confidence Level *

Factor Description	Low-Value High-Value	Relative Change in Oil Consumption 700 RPM Low Idle	1800 RPM No Load	1800 RPM Rated Pwr	1600 RPM 213 HP
Top Ring Gap Orifice Area	1.8E-4 in2 5.8E-4 in2	+4*	0	0	-1
Second Ring Orifice Area	4.5E-4 in2 14.0E-4 in2	+4*	0	0	0
Interring Vol Top/2nd rings	0.079 in3 0.158 in3	0	0	+5**	+5**
Interring Vol 2nd/3rd rings	0.57 in3 .114 in3	0	0	+2.5	+1
Second Ring Mass	.066 lb. .082 lb.	-1	+5*	+2.5	+4*
Side Seal Position Top Ring	Slight Front Edge Heavy Front Edge	+1	+5*	+2.5	-4*
Side Seal Position 2nd Ring	Flat Seal Rear Edge	+1	0	0	-1

Relative Oil Consumption Weighting Factor Scale

Unacceptable Acceptable Preferred
0----1----2----3----4----5----6----7----8----9----10

Oil film thickness and ring/liner void area predictions pointed to the fact that liner distortion on a typical L10 engine is a detriment to oil control. Overall, the model proved to be an interesting and important tool in designing and developing cylinder kit components. It is planned to make further use of the tool in Phase 2 and 3 of the program as the design details are optimized.

VIII. SUBCONTRACTOR EFFORTS

A. RING COATING ANALYSIS AT UTRC

To help guide the development of advanced wear resistant coatings for piston rings, an analytical study was performed of various coating alternatives. This study was subcontracted to United Technologies Research Center due to their experience in designing and analyzing coatings for aircraft engines. Principal investigators at UTRC were T. P. Slavin, R. P. Huston, and M. Bak.

The Phase I effort was focussed on developing simplified thermal/structural models using finite element methods to predict the stress state of candidate wear resistant coating systems on diesel engine piston rings during engine service. The primary goal of this program was to define a preliminary ranking of the potential performance of each candidate coating and ring material after coating deposition, during installation of the ring and during L10 diesel engine operating conditions. The thermal/structural model has been defined for the L10 diesel engine service cycle based upon thermal and mechanical boundary condition information provided by Cummins Engine Company, Inc. During the contract period, the following accomplishments were achieved:

Developed a finite element thermal/structural model of a simple L10 piston ring geometry with a 0.010 in. coating present on a beveled edge along the ring outer diameter. This model is capable of being expanded to accommodate both coating thickness and geometry variations (such as a grooved ring for inlaid coatings) for additional analyses.

Utilized the model as a screening method to predict the following for each candidate coating system and subsequently rank the predicted performance of each candidate coating:

- as-deposited room temperature residual stresses
- stresses generated during ring installation
- thermal gradient profile of the coated rings during L10 engine low idle and maximum power operating condition
- thermal/mechanical stresses present in the coated ring during L10 engine maximum power/maximum cylinder pressure operating condition

All predicted stresses from the finite element analyses were dependent upon the assumed value for the coating/substrate Stress-Free Temperature (SFT). The SFT is the temperature at which no stresses are present between the coating and the substrate and is usually associated with the coating deposition temperature.

The ring coating candidate materials in this study were:

Coating	Material Class
- Cr_2O_3	Ceramic
- ZrO_2-TiO_2-Y_2O_3 (ZTY)	Ceramic
- Cr_3C_2-NiCr	Cermet
- NiCrBSi	Metal
- Cr plate (baseline)	Metal

The ring substrate materials in this study were:

- Ductile Iron (DI) (baseline)
- Inconel 625 (IN)
- 422 Stainless Steel (SS)

The highlights from the finite element analyses were:

All of the candidate thermally sprayed coatings as well as the baseline Cr plate on ductile iron rings were predicted to be capable of being manufactured and operated in a diesel engine without coating failure.

Although these coating/ring combinations were not modeled in this study, engineering judgement dictates that it is likely that the predicted operational stress states for Cr_2O_3, ZTY and Cr_3C_2-NiCr coatings would be generally neutral if the coatings were present on the 422 stainless steel instead of the baseline ductile iron ring material. The small thermal expansion coefficient mismatches between these candidates and the 422 stainless steel material would limit the generated stress levels resulting from temperature changes, thus making these coating/ring combinations potentially successful during diesel engine service. Coating Rankings Based Upon Predicted Stresses: The general ranking of the candidate coating systems was variable depending upon the process stage modeled for both manufacturing and operation. The general criteria for ranking the coating systems was that a neutral to slightly compressive predicted stress state exist for the coating to be highly ranked. Tensile and high magnitude compressive predicted stress states were considered to be less desirable for the coating systems. The overall judgement in each instance was based on both the predicted stress data and engineering judgement.

A. Diesel Engine Operational Stresses:

* The general ranking of the candidate coating materials was based upon the bulk (not localized) predicted operational stress states. The differences between the candidates in this category were not substantial.

1. NiCrBSi on DI, IN and SS rings.
2. ZTY, Cr_2O_3 and Cr_3C_2-NiCr on DI rings.
3. Cr plate on DI rings.

** The physical constraints imposed by the ring groove and cylinder wall in the finite element model distort the ring/coating interfacial stress field into an "S-shape" (in a static model) and thus result in the generation of enhanced maximum/minimum predicted localized operational stresses which may cause local yielding/microcracking in both the coatings and the ring materials. The Cr plate on DI rings may be the best candidate modeled with regard to the localized operational stresses due to its low assumed SFT value.

** All of the coating candidates potentially may be predicted to successfully withstand the localized stresses predicted for engine operation if lower SFT values are assumed.

** The major fraction of the predicted coating and ring stresses during L10 engine maximum power operation were the result of the thermal conditions (temperature differences vs. SFT and thermal expansion mismatches). The mechanical contribution (cylinder gas pressure) to the predicted operational stress state was small as compared to the thermal contribution.

B. As-Deposited Residual Stresses:

Figure 63 shows a schematic of the coated piston ring cross section including the locations of as-deposited, room temperature residual stresses.

* The general ranking of the candidate coating materials as based upon the predicted as-deposited, room temperature residual stress states:

1. NiCrBSi on DI and IN rings.
2. Cr plating on DI rings.
3. Cr_3C_2-NiCr on DI rings.
4. Cr_2O_3 on DI rings.
5. ZTY on DI rings.
6. NiCrBSi on SS rings.

** All of the candidate coating materials would be unlikely to fail due to the predicted as-deposited residual stresses.

** The maximum predicted room temperature, as-deposited residual stresses were compressive for three candidate thermal spray coatings (Cr_2O_3, ZTY and Cr_3C_2-NiCr) on ductile iron rings. The NiCrBSi coating had a neutral predicted residual stress state on both ductile iron and Inconel 625 rings but had a tensile predicted residual stress state on the 422 stainless steel ring material. Cr plate at an SFT=RT had zero predicted stresses while at an SFT=200F the predicted stresses were slightly compressive.

C. Ring Installation Stresses:

* The general ranking of the candidate coating materials based upon the predicted stresses for ring installation including the room temperature residual stresses was as follows:

Closing the Ring for Installation into Cylinder

1. ZTY, Cr_2O_3, Cr_3C_2-NiCr on DI
2. NiCrBSi on DI, IN rings.
3. Cr plate on DI rings.
4. NiCrBSi on SS rings.

** For installation of the coated rings onto the piston crown and subsequently into the cylinder, the ceramic candidates Cr_2O_3 and ZTY were predicted to have net compressive circumferential stresses while the remaining candidates were predicted to have net tensile circumferential stresses.

Thermal Analysis

Thermal analysis by finite element methods determined that there was no significant difference in the temperature distribution of any of the candidate coating and ring materials during both L10 engine low idle and maximum power operation except for ZTY. The ZTY coating material was predicted to act as a thermal barrier and inhibit heat transfer from the piston ring into the cylinder wall during diesel engine operation.

Overall Coating Assessment

All four of the thermal spray coating materials (ceramic, metal and cermet classes) modeled on ductile iron rings have been predicted to possess the strength and temperature capability to operate as an alternative to the baseline Cr plate in the diesel engine environment examined in this study. It is recommended that each candidate coating should be engine tested and analyzed for actual engine performance capability in order to verify the model's predictions. In addition, the wear resistance of the coating candidates should be documented in both rig and engine tests to assess the coatings' wear capability for comparison with the coatings' properties and predicted stress states. The best predicted candidate coating/substrate systems, based upon both the model and engineering judgement, are as follows:

1. NiCrBSi on DI and IN rings.
2. ZTY, Cr_2O_3 and Cr_3C_2-NiCr on DI rings.
3. Cr_3C_2-NiCr, ZTY and Cr_2O_3 on SS rings (engineering judgement).

B. CYLINDER KIT FRICTION MEASUREMENT AT THE UNIVERSITY OF MICHIGAN

The engine research lab at the University of Michigan has previously developed a method of instantaneous measurement of piston and ring friction using what they had termed the "instantaneous IMEP method". This method entails instantaneous measurement of engine rpm, cylinder pressure, and connecting rod force along with calculation of inertia force and pressure force. When these forces are summed, the resultant is the friction between the piston/rings and the cylinder liner. As might be imagined, accurate measurements are critical since large values are being difference to produce a small result.

The instantaneous IMEP method was applied to an L10 engine for this program. This required addition of a cylinder pressure transducer, connecting rod strain gauges, shaft encoder, and a "grasshopper linkage" for running instrumentation wires out of the crankcase. A suitable PC based data acquisition system was used to record the data, perform the calculations, and plot the results. At the conclusion of Phase 1 of this program, the engine has been installed in the test cell, all instrumentation installed and running, and system check-out underway. The test plan includes:

- Varying engine speed and load
- Varying oil temperature
- Comparing aluminum and articulated pistons
- Varying the amount of piston cooling oil flow

IX. PHASE 2 PLAN

The objectives of Phase 2 of the In-Cylinder Components program are the following:

1. Perform detailed design and analysis of all proposed component concepts.

2. Develop special manufacturing techniques where required.
3. Bench and rig tests to refine designs.

A partial list of the necessary tasks follows.

PISTON

- Detailed design of the spherical joint piston
- Manufacturing process development of squeeze cast fiber reinforcement, ring groove insert, and socket machining
- Spherical joint analysis (loading lubrication)
- Ring groove insert wear test
- Plasma spray fatigue test
- Hydropulse fatigue test

CONNECTING ROD

- Detailed design and analysis
- Ball joint machining development
- Fatigue test

CYLINDER HEAD

- Detailed design and analysis
- Process development of combustion face insert, plasma spray insulation, ceramic port insulation
- Thermal rig test
- Oil cooling enhancement

CYLINDER LINER

- Detailed design and analysis
- Test and development of radial combustion seal
- Surface treatment development (wear coatings)
- Turbulator

PISTON RINGS

- Design and analysis of higher strength and conformability
- Ring fatigue tests
- Wear testing of rings, grooves, and liner

The overall goal of the Phase 2 effort is to have designs and prototype components ready for engine testing in Phase 3. Mechanical, performance, and emissions testing will then take place on a single cylinder engine to prove out the concepts now being designed.

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In order to access world class technology in certain specialized areas, Cummins has sub-contracted a portion of the technical effort. In all sub-contract cases there has been close interaction with one or more of the core group to assure that the emerging technologies and developments are brought into Cummins. The major sub-contracts and principle contacts have been:

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16. Abstract

In order to achieve the Heavy Duty Transport Technology (HDTT) program goal of .250 lb/bhp-hr brake specific fuel consumption, an engine system was proposed consisting of two stage turbocharging (to enable high BMEP at low engine speed), turbocompounding, low intake manifold temperature, low heat rejection, and high fuel injection and peak cylinder pressures. All components and systems are based on the Cummins L10 engine, a production engine of 10 litre displacement of modern design and proven performance. Thermal analysis of the cylinder head and liner determined that by using strategically placed oil cooling with suitable enhancement, it is possible to achieve acceptable component temperatures and thermal fatigue life while significantly reducing the overall system size and complexity by eliminating cooling water. A cylinder head concept using oil cooled grey iron, cast-in ceramic intake and exhaust ports, a high temperature metal combustion face insert, and plasma sprayed insulation on part of the combustion face is being pursued. The key in-cylinder component which has received much attention in this first phase is the piston. The concept being refined consists of a spherical piston/connecting rod joint for high load carrying capabilities and uniform thermal loads, ceramic fiber reinforced aluminum for light weight and high strength, and a combustion chamber thermal barrier coating of plasma sprayed zirconia.

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